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# Design and test of a rocket motor expansion joint

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California Institute of Technology

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**CALIFORNIA INSTITUTE OF TECHNOLOGY**

# DESIGN AND TEST OF A ROCKET MOTOR EXPANSION JOINT

Thesis by

Lt. Cmdr. R. L. Reiserer U.S.N.
Lt. K. P. Barden

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# DESIGN AND TEST OF A ROCKET MOTOR

# EXPANSION JOINT

Thesis by

Lt. Cmdr. R. L. Reiserer U.S.N.
Lt. K. P. Barden U.S.N.

In Partial Fulfillment of the Requirements for
the Professional Degree in Aeronautical Engineering.

California Institute of Technology

Pasadena, California

1947

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## SUMMARY

The purpose of this investigation was to carry on the work of investigating stresses encountered in the design of rocket motor expansion joints. The previous work was done on a semicircular expansion joint by Commander N. J. Kleiss, U.S.N., and Lt. Comdr. S. W. Kerkering, U.S.N. in a thesis submitted to the Aeronautics Department of the California Institute of Technology.

Information was desired to make possible a more favorable design. The problem consists of continuing the investigation on a different design, thereby determining a trend for future designs.

The design chosen was a semicircular ring shell with reversed curvatures of the edges. It was felt that this configuration would eliminate most of the bending stresses found present at the weld of the semicircular design, and would make possible a more even distribution of stresses throughout the expansion joint.

Because the stresses encountered were so far beyond the elastic limit of the material, even for small total elongations of the expansion joint, no theoretical solutions of the problem were possible.

An analysis of the curves and data shows the joint to be unsatisfactory, particularly from the viewpoint of withstanding the internal pressure. Since the previous work indicated that the internal pressure was not critical for the semicircular design, it seems that a compromise of the two designs would more satisfactorily meet the requirements of withstanding both

the elongation and the internal pressure.

An alternate design has been indicated which will make possible a trend of designs that should lead to a solution.

## INTRODUCTION

If the science of guided missiles is to continue to advance, each component part of the missile must be perfected. This means that a satisfactory propulsive system capable of continual operation to some degree must be developed. The development of a satisfactory expansion joint for the rocket motor is one phase of this problem.

The expansion joint must withstand internal pressures up to 600 p.s.i., and must be capable of large repeated elongations. The difference in temperature between the inner and outer shells causes relative expansions and contractions which must be absorbed by an expansion joint. The external dimensions of the expansion joint are usually dictated by aerodynamic considerations, while the internal dimensions are determined to a great extent by the fuel flow requirements. Further considerations in design are weight, simplicity of design, ease of production, and maintenance.

Figure A shows the combustion chamber, cooling chamber, and expansion joint of a typical rocket motor. Fuel under high pressure is pumped through the cooling chamber and acts as a coolant for the combustion chamber. High combustion temperatures cause elongation of both the inner and outer shells. However, because the temperature of the inner shell is several hundred degrees higher than that of the outer shell, the elongations are not equal. Consequently, an expansion joint that will allow the outer shell to expand with the inner shell without imposing too

severe loads on the latter is required. The elongation of the rocket motor with the temperature distribution as shown in Figure A is of the order of 0.10" in ten inches. This indicates that for a motor of a given length two or three expansion joints may be required.

The tests on the specimens were planned to give: a) the stresses at critical positions on the expansion joint at various internal pressures and elongations, b) the elongation and loadings which would give permanent set, and c) the number of repeated elongations necessary to cause failure.

Four specimens of the same design but of various thicknesses were tested. The first two specimens were 0.028 and 0.035 inches thick respectively. The third and fourth were both from 0.053 to 0.057" thick. The specimens were manufactured by a spinning process in which a piece of flat sheet was spun down over a wooden mandrel while hot. This made accurate control of the thicknesses almost impossible. It was originally intended that two of the specimens be 0.04" and two be 0.05" thick in order that comparisons could be made between this and the semicircular design of Ref. 4. However, during the fabrication of the first set of specimens, the standard sheet stock used (0.0418 and 0.0538) was decreased in thickness so much that the first two specimens mounted were useless except as introductions to the problems ahead and as preliminary indications of where the critical stresses occured.

## II

## EQUIPMENT AND TEST PROCEDURE

Fig. B illustrates the method of assembling the expansion joint on the pressure vessel. The entire assembly was manufactured of mild steel (1020) and assembled by welding. The first specimen was 0.028" in thickness, the second 0.035, and the third and fourth each from 0.053 to 0.057. The desired internal pressures were obtained by pumping oil through a 5/8" pipe threaded hole at one end of the cylinder. A Blackhawk hydraulic jack was used to supply the pressure. Pressure readings were taken from a large, dial type, N.A.C.A., hydraulic gage.

Depending on whether the specimen was in tension or compression, either two steel bars or two steel plugs were screwed into the centers of the top and bottom plates of the pressure vessel. (See Photograph #3).

A Tinius Olsen beam balance testing machine was used to apply the tension or compression and to control the elongation. (See Photographs 1 and 2.). The force measuring components of this machine were very good, however trouble was encountered in trying to obtain a given elongation at a constant internal pressure in the specimen.

Spherical bearings were used in tension tests and ball bearing pyramids in compression tests in order to apply a centralized load without end moments.

Values of stress were obtained by means of SR-4, type A-8, Baldwin Locomotive electric wire resistance strain gages. Six gages were mounted on the first specimen and eleven on the

second specimen. The locations of attachment are illustrated in Fig. C. On specimen III, eight gages were used, and on specimen IV ten gages were applied. Their locations are also shown in Figure C. On all specimens the gages were located radially at 90° quadrants in order to obtain average readings.

The strain gages were used in conjunction with a multiple channel Wheatstone Bridge designed and made at the California Institute of Technology. Voltage measurement was made by a Leeds and Northrop Potentiometer. This apparatus was capable of measuring the change in voltage in the strain gages to an accuracy of 0.001-millivolt, which corresponds to an accuracy of 12.95 P.S.I. of stress or 0.037% of the proportional limit strain.

The over-all elongation of the specimen was measured by means of a dial gage mounted between the working platforms of the Tinius Olsen machine. This gage measured elongations within 0.001" accuracy.

The first expansion joint specimen tested revealed the need for strain gage calibration well beyond the elastic limit of the material. Therefore, a curve of stress vs. millivolts was obtained in the following manner.

Standard 0.5", flat, pin-ended tensile test specimens were made of the same material used in the manufacture of the expansion joints. Electric strain gages were mounted on opposite sides of the specimens so as to eliminate the effects of any bending. These tensile test specimens were used to obtain the data and curve shown in Table 1 and Figure 1 respectively.

This curve made it possible to convert all millivolt

readings directly to stress values without the use of gage constants. These stresses were used for the final plotting of $\sigma$ vs. $\delta$.

Specimen I (thickness = 0.028") was too thin to withstand the required internal pressure and was also poorly welded. This specimen was loaded only in tension and was used primarily to locate the regions of highest stress. No data herein presented.

Before mounting Specimen II in the testing machine it was pressure tested by means of water and air pressure, and all leaks repaired. Data taken during runs with this specimen indicated that the 90 p.s.i. used during the pressure testing had caused permanent set in the specimen. Therefore this specimen, since also obviously too thin, was used to determine the position of highest stresses in the expansion joint with internal pressure. No data or curves for the first two specimens are included in this report.

Specimen III (t = 0.053 - 0.057) was first mounted in tensile load only. The specimen was repeatedly elongated from zero to 0.10" for 25 cycles with no internal pressure. Data for this test is recorded in Table 2a, 2b, 2c, 2d. The curves of stress vs. elongation are plotted for the 1st and the 25th cycle in Figures 2 to 12. No data or curves of the intermediate cycles are presented because of the similarity of results of all cycles.

With various constant internal pressures of from 50 - 400 p.s.i. applied, this specimen was then elongated until the strain gages readings indicated the presence of stresses equal to the

highest of those obtained during the 25 cycle test. In these tests the specimen was placed in compression to control the elongation. Finally an attempt was made to vary the elongation with an internal pressure of 600 p.s.i. However, the specimen failed at the weld before 600 p.s.i. was obtained. ($\delta = 0$). The stresses were read from the calibration curve and recorded in Table 3a, 3b, 3c, 3d. Curves are plotted in Figures 13 to 18.

The procedure and treatment of the tests on specimen IV (t = 0.053 - 0.057) were the same as those for No. III except that internal pressure was used in all tests. Pressure varied from zero to 500 p.s.i. This specimen withstood the 600 p.s.i. internal pressure for only one half of a cycle (from zero to 0.10") before failure at the weld. Data and corresponding curves are shown in Table 4a, 4b, 4c, 4d and Figures 19 to 24.

## III

## DISCUSSION OF RESULTS

In both the semicircular design discussed in Ref. 4 and the design with the turned up edges studied in this report, stresses beyond the elastic limit of the material were encountered at very small total elongations. In the design with the turned up edges the internal pressure was more critical than it was in the semicircular joint and the amount of deformation increased very appreciably with increasing internal pressure. Curves in Figures 13 to 24 are plots of stress vs. elongation at constant pressures and show the wide variation of stresses encountered as the pressure was increased.

Figs. 2 to 5 show that, with no internal pressure, elongations of 0.045" were obtained with no appreciable permanent set. However, as the pressure was increased, permanent set was noticeable at increasingly smaller elongations. Above 200 p.s.i. stresses beyond the yield were observed even at zero elongation.

In order to evaluate the separate effects of elongation and pressure, the first 0.055" thick specimen (No. III) was run through twenty-five repeated cycles of elongation from zero to 0.10" without any internal pressure. These twenty-five cycles were completed without failure but with a permanent set of 0.020" in total elongation.

Figures 6 to 12 show that after 25 repeated cycles, the stresses encountered during expansion are essentially the same as those found during the initial expansion except for the 0.020" permanent set. The 0.020" permanent set was present after the 1st cycle but remained constant throughout the remainder of the repeated

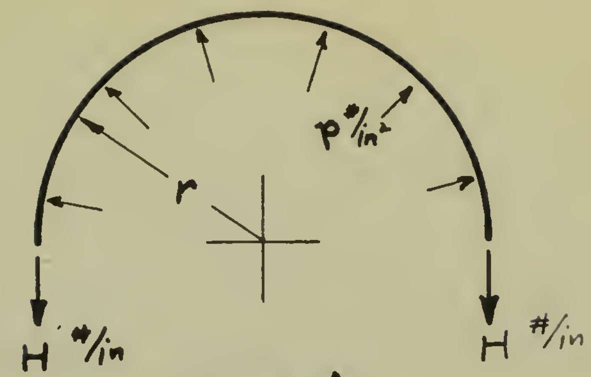
loading test. This indicated that the specimen was capable of many more cycles under the same conditions.

As internal pressure was applied, it was assumed that no failure would occur if the strain gage readings were kept below the readings recorded during the repeated load tests.

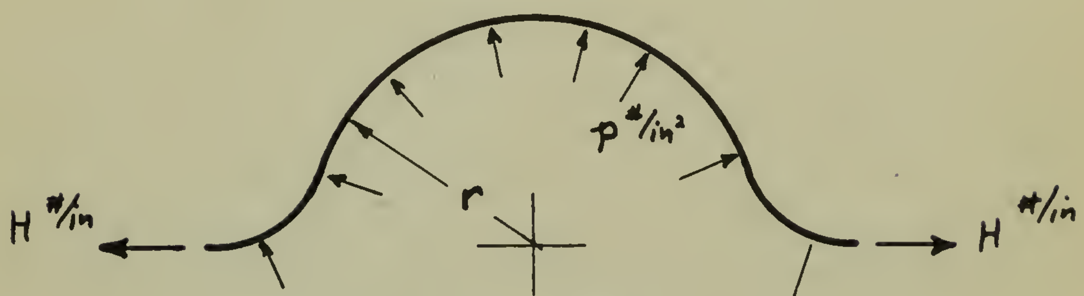
On this assumption and a comparison of the curves for specimen III at various internal pressures, the following chart was developed. This chart gives an indication of the maximum total elongation possible for twenty-five cycles at different constant pressures.

| $P$ #/in$^2$ | $\delta$ in. |
|---|---|
| 0 | 0.10 |
| 50 | 0.10 |
| 100 | 0.080 |
| 150 | 0.074 |
| 200 | 0.070 |
| 300 | 0.050 |
| 400 | 0.020 |
| 500 | ---- |
| 600 | 0 |

Failure occured in this specimen before 600 p.s.i. was reached even at zero deflection. Since the semicircular design was capable of two and one-half cycles from zero to 0.20" with 600 p.s.i., the semicircular expansion joint is more capable of withstanding the high internal pressure. The reason for this is most clearly demonstrated in the accompanying sketch which shows that in Case A membrane stresses in a radial direction are more suitable for withstanding the pressure loads than they are in Case B.




CASE A




CASE B

In Case B large membrane stresses in both the circumferential and the axial directions would be required to maintain an equilibrium of forces, since neither of these membrane stresses is capable of contributing much resistance to forces normal to the surface of the expansion joint near the weld. Experimentally, this seemed to be verified since Case A was capable of withstanding 600 p.s.i. in two thicknesses (0.04" and 0.05") while Case B failed before 600 p.s.i. was reached even though the specimen was made of thicker material. (t = 0.055").

Strain gages mounted at the outermost circumference of the fourth specimen showed that the stresses present at this part of the expansion shell were negative (compression). These stresses remained practically constant as the elongation was varied and the internal pressure was held at given constant values. This

indicates the presence of a bending point of inflection. (See Figures 23 and 24). The combination of the radial contraction of the joint and the apparent bending action results in compressive components in both directions in this region of the joint. In order to better illustrate the stress distribution, both bending and membrane stresses at various points are shown in the accompanying schematic diagram. (See Figure d.)

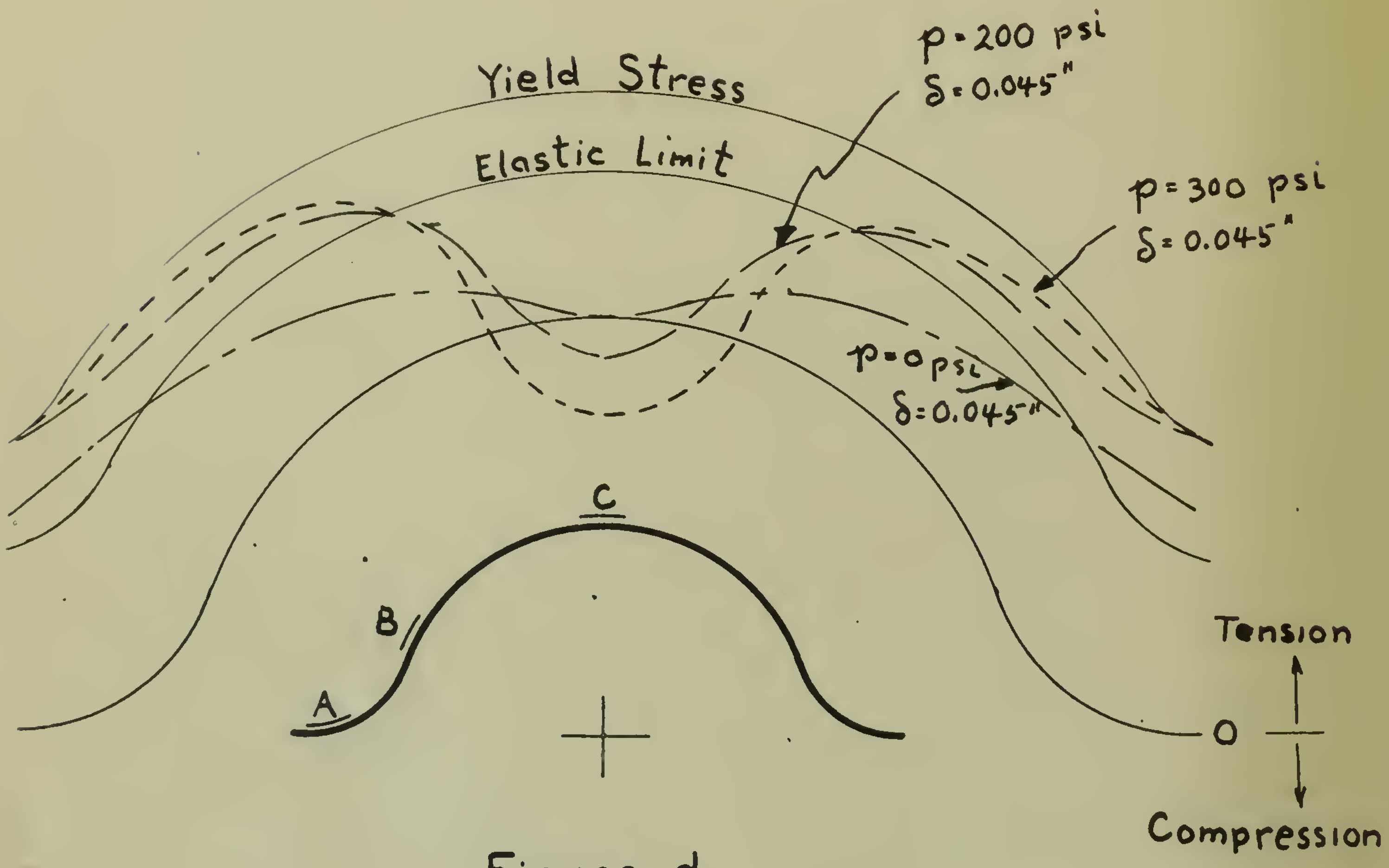
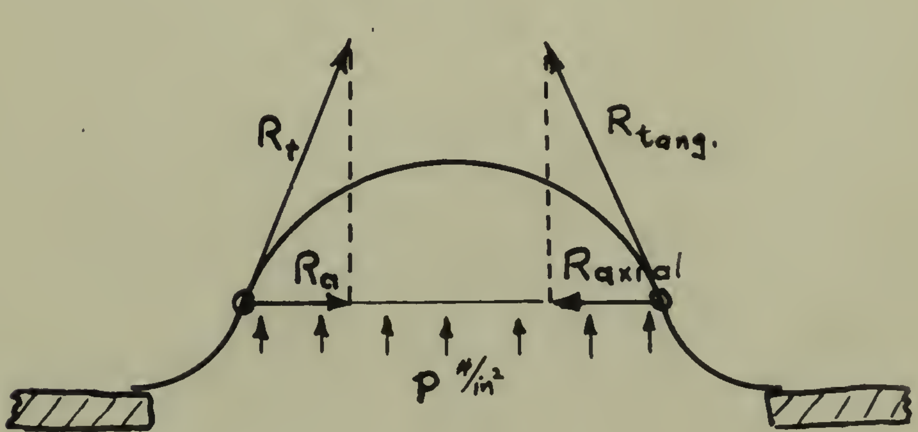



Figure d

An attempt was made to analyse the effect of internal pressure on the spring constant of the expansion joint. However, because the material of the joint exceeded the elastic limit at such small elongations and because the joint changes shape under load the problem is non linear and not easy to discuss. In order to best illustrate the behavior of the spring constant under pressure, curves of the spring constant K are plotted against the elongation of the joint at several internal pressures. (See Fig. 25). The spring constant for these curves was obtained by dividing the difference between the measured axial compressive force and the calculated axial tensile force due to internal pressure by the elongation of the expansion joint.

Over the range of pressures and elongations studied, the spring constant was increased at the small elongations with increases in internal pressure. At elongations close to 0.10", the spring constant at all pressures tended to converge with the spring constant of the expansion ring without internal pressure. This action is probably due to the axial components of the membrane forces ~~that~~ which were developed in the ring by the high internal pressures. (See accompanying figure).

Figure e



On the basis of these tests the expansion joint with the turned up edges was found to be unsatisfactory for the following reasons.

1. The design was incapable of withstanding the design internal pressure of 600 p.s.i.

2. Permanent set was obtained with relatively low internal pressures at elongations of from 0.04 to 0.06".

3. Stresses were the greatest close to the weld where the physical properties of the material were the poorest.

4. The spring constant of the joint increases at high internal pressures.

The design with the turned up edges appeared capable of withstanding greater elongations than the semicircular design as long as the internal pressure was held below 300 p.s.i. Above 300 p.s.i. the semicircular design appeared to be the better of the two.

# IV

## RECOMMENDATION FOR FUTURE STUDIES

It is believed that a more suitable design is possible by combining the advantages of both of the types studied to date. Fig. f shows a suggested design which should combine these advantages.

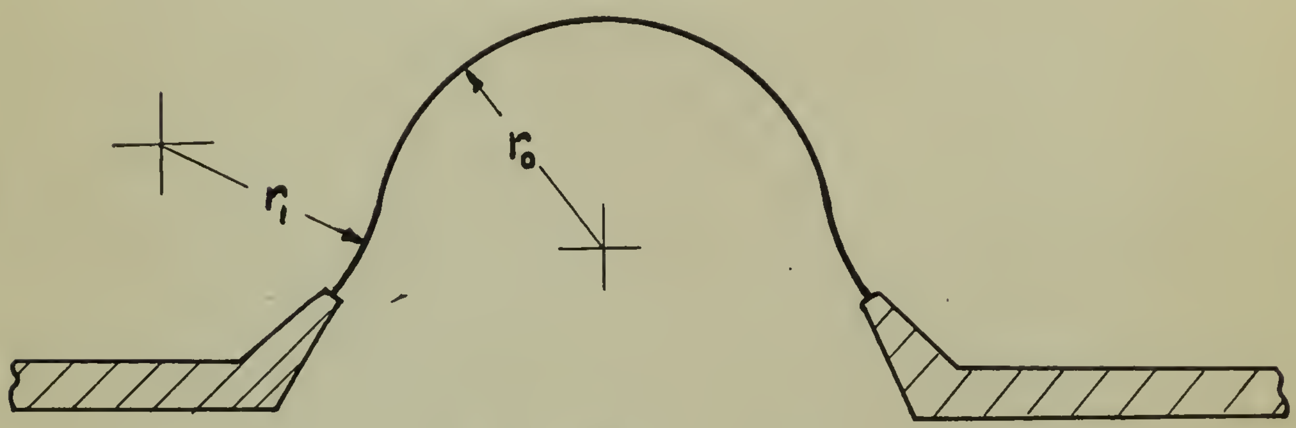



Figure f

This configuration would make possible the use of the lightest gauge material which would still be capable of withstanding the 600 p.s.i. internal pressure. At the same time it should keep the total force in the axial direction required to elongate the specimen a minimum. This design should also eliminate most of the extreme bending stresses that occured at the weld of the semi-circular design.

It is further recommended that a more thorough study be made of the effect of the circumferential or hoop stresses on the rigidity of the expansion joint.

V

## REFERENCES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3. Theory of Elasticity — S. Timoshenko
4. Some Studies of Expansion Rings in Rocket Motors — Cmdr. N. J. Kleiss, U.S.N. Lt.Cmdr. S. W. Kerkering, U.S.N.

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TABLE 1

Stress-Millivolt Curve

Area = 0.061 x 0.50 = 0.0305 sq. in.

| Load | Gage 1 | Gage 2 | Gage 3 | $\Delta v$ 1 | $\Delta v$ 2 | $\Delta v$ 3 | Ave. | Stress $\sigma$ |
|---|---|---|---|---|---|---|---|---|
| 50 | .063 | .268 | .256 | .063 | .268 | .256 | .196 | 1640 |
| 100 | .250 | .379 | .387 | .187 | .111 | .131 | .339 | 3280 |
| 150 | .420 | .488 | .508 | .170 | .109 | .121 | .472 | 4920 |
| 200 | .615 | .615 | .631 | .195 | .127 | .123 | .620 | 6560 |
| 250 | .791 | .740 | .753 | .176 | .125 | .122 | .762 | 8200 |
| 300 | .960 | .861 | .879 | .169 | .121 | .126 | .900 | 9840 |
| 350 | 1.097 | .992 | .995 | .137 | .131 | .116 | 1.060 | 11480 |
| 400 | 1.266 | 1.097 | 1.100 | .105 | .105 | .169 | 1.155 | 13120 |
| 450 | 1,432 | 1.237 | 1.237 | .144 | .140 | .137 | 1.302 | 14760 |
| 500 | 1.592 | 1.372 | 1.359 | .160 | .135 | .122 | 1.441 | 16400 |
| 550 | 1.753 | 1.510 | 1.487 | .161 | .138 | .128 | 1.583 | 18010 |
| 600 | 1.912 | 1.648 | 1.615 | .159 | .138 | .128 | 1.725 | 19680 |
| 650 | 2077 | 1.783 | 1.739 | .165 | .135 | .124 | 1.866 | 21310 |
| 700 | 2.233 | 1.930 | 1.873 | .156 | .147 | .134 | 2.012 | 22950 |
| 750 | 2.390 | 2.070 | 2.005 | .157 | .140 | .132 | 2.155 | 24600 |
| 800 | 2.550 | 2.213 | 2.130 | .160 | .143 | .125 | 2.298 | 26210 |
| 850 | 2.708 | 2.359 | 2.266 | .158 | .146 | .136 | 2.444 | 27900 |
| 900 | 2.863 | 2.506 | 2.392 | .155 | .147 | .124 | 2.587 | 29500 |
| 950 | 3.014 | 2.653 | 2.520 | .151 | .147 | .128 | 2.729 | 31900 |
| 1000 | 3.171 | 2.797 | 2.653 | .157 | .144 | .133 | 2.874 | 32800 |
| 1050 | 3.322 | 2.949 | 2.787 | .151 | .152 | .134 | 3.019 | 34500 |
| 1100 | 3.470 | 3.090 | 2.918 | ..48 | .141 | .131 | 3.159 | 36100 |
| 1150 | 3.618 | 3.238 | 3.049 | .148 | .148 | .131 | 3.302 | 37700 |
| 890 | 5.015 | | | 1.397 | | | 5.015 | 29200 |
| 908 | 10.423 | Gages went | | 5.480 | | | 10.423 | 29800 |
| 933 | 11.080 | bad. | | .653 | | | 11.080 | 30600 |
| 1000 | 11.715 | | | .635 | | | 11.715 | 32800 |
| 1034 | 12.710 | | | .995 | | | 12710 | 33930 |
| 1036 | 13.163 | | | .453 | | | 13.163 | 33950 |

All Gage readings are in Millivolts

and all Stresses are in lbs/sq. in.

TABLE 2a

Specimen III

t = 0.055"    No Internal Pressure
1st Cycle    Tension Only

| Load | x $10^{-3}$ | Millivolts Gage 1 | 2 | 3 | 4 | 5 | 7 | 6 |
|---|---|---|---|---|---|---|---|---|
| 748 | 5 | .193 | .250 | .378 | .307 | .083 | .037 | .175 |
| 1947 | 10 | .657 | .885 | .981 | .912 | .400 | .241 | .563 |
| 3068 | 15 | .969 | 1.347 | 1.464 | 1.408 | .562 | .413 | .871 |
| 4327 | 20 | 1.367 | 1.886 | 1.967 | 1.908 | .750 | .512 | 1.192 |
| 5293 | 25 | 1.582 | 2.197 | 2.323 | 2.278 | .871 | .545 | 1.386 |
| 6241 | 30 | 1.879 | 2.625 | 2.751 | 2.622 | 1.035 | .631 | 1.664 |
| 7227 | 35 | 2.110 | 2.945 | 3.092 | 3.011 | 1.198 | .728 | 1.910 |
| 8265 | 40 | 2.419 | 3.369 | 3.522 | 3.391 | 1.357 | .806 | 2.177 |
| 9330 | 45 | 2.708 | 3.757 | 3.929 | 3.814 | 1.510 | .933 | 2.484 |
| 10272 | 50 | 3.013 | 4.142 | 4.234 | 4.165 | 1.648 | .989 | 2.694 |
| 11382 | 55 | 3.265 | 4.527 | 4.708 | 4.591 | 1.813 | 1.063 | 3.010 |
| 12305 | 60 | 3.539 | 4.956 | 5.052 | 4.939 | 1.957 | 1.148 | 3.200 |
| 13417 | 65 | 3.800 | 5.392 | 5.497 | 5.345 | 2.137 | 1.241 | 3.549 |
| 14352 | 70 | 4.060 | 5.838 | 5.878 | 5.644 | 2.259 | 1.307 | 3.771 |
| 15320 | 75 | 4.277 | 6.252 | 6.223 | 5.975 | 2.452 | 1.394 | 4.078 |
| 16265 | 80 | 4.560 | 6.671 | 6.544 | 6.259 | 2.572 | 1.490 | 4.311 |
| 16925 | 85 | 4.758 | 7.041 | 6.904 | 6.497 | 2.740 | 1.625 | 4.588 |
| 17565 | 90 | 5.050 | 7.411 | 7.203 | 6.720 | 2.901 | 1.802 | 4.818 |
| 18025 | 95 | 5.318 | 7.657 | 7.359 | 6.849 | 3.105 | 2.068 | 5.088 |
| 18400 | 100 | 5.436 | 8.003 | 7.552 | 7.011 | 3.333 | 2.335 | 5.282 |

TABLE 2b

Specimen III

| Load lbs | x 10$^{-3}$" δ | Stresses psi, Gage 1 | 2 | 3 | 4 | 5 | 7 | 6 |
|---|---|---|---|---|---|---|---|---|
| 748 | 5 | 1800 | 2800 | 4200 | 3250 | 800 | 200 | 1750 |
| 1947 | 10 | 7300 | 9950 | 11100 | 10200 | 4450 | 2600 | 6200 |
| 3068 | 15 | 10900 | 15150 | 16400 | 15900 | 6200 | 4600 | 9800 |
| 4327 | 20 | 15350 | 21350 | 22250 | 21500 | 8400 | 5700 | 13400 |
| 5293 | 25 | 17800 | 24800 | 26200 | 25800 | 9800 | 6000 | 15800 |
| 6241 | 30 | 21300 | 29700 | 31100 | 29600 | 11500 | 7100 | 18700 |
| 7227 | 35 | 23800 | 36200 | 35000 | 33900 | 13400 | 8100 | 21600 |
| 8265 | 40 | 27250 | 38000 | 38350 | 38200 | 15250 | 8500 | 24500 |
| 9330 | 45 | 30600 | 38100 | 36900 | 37750 | 17000 | 10400 | 28100 |
| 10272 | 50 | 34000 | 35100 | 34000 | 35000 | 18500 | 11100 | 30400 |
| 11382 | 55 | 36975 | 32300 | 31100 | 31850 | 20450 | 11800 | 34000 |
| 12305 | 60 | 38290 | 29900 | 29400 | 29900 | 22200 | 12800 | 36200 |
| 13417 | 65 | 37800 | 28050 | 27750 | 28250 | 24000 | 14000 | 38350 |
| 14352 | 70 | 36000 | 26850 | 26700 | 27300 | 25500 | 14700 | 37950 |
| 15320 | 75 | 34200 | 25950 | 25950 | 26450 | 27700 | 15750 | 35800 |
| 16265 | 80 | 32100 | 25400 | 25600 | 25950 | 29000 | 16800 | 33750 |
| 16925 | 85 | 30900 | 25300 | 25350 | 25600 | 31000 | 18300 | 31900 |
| 17565 | 90 | 29400 | 25350 | 25250 | 25400 | 32800 | 20300 | 30550 |
| 18025 | 95 | 28350 | 25400 | 25300 | 25350 | 35100 | 23400 | 29300 |
| 18400 | 100 | 27975 | 25550 | 25350 | 25300 | 37700 | 26350 | 28500 |

TABLE 2c

Specimen III

t = 0.055"
25th Cycle

No Internal Pressure
Tension Only

| Load lbs | x $10^{-3}$ $\delta$" | Gage Millivolts 1 | 2 | 3 | 4 | 5 | 7 | 6 |
|---|---|---|---|---|---|---|---|---|
| 0 | 26 | - .180 | .880 | .180 | -.480 | .490 | .802 | .424 |
| 3260 | 40 | .835 | 2.340 | 1.590 | .896 | 1.070 | 1.150 | 1.540 |
| 7345 | 60 | 2.240 | 4.160 | 3.360 | 2.600 | 1.770 | 1.535 | 2.710 |
| 11875 | 80 | 3.620 | 6.010 | 5.160 | 4.410 | 2.490 | 1.952 | 4.014 |
| 16235 | 100 | 5.075 | 7.904 | 6.922 | 6.145 | 3.140 | 2.302 | 5.100 |

TABLE 2d

| Load lbs | x $10^{-3}$ $\delta$" | Gage Stresses 1 | 2 | 3 | 4 | 5 | 7 | 6 |
|---|---|---|---|---|---|---|---|---|
| 0 | 26 | -1800 | 9900 | 1800 | -5200 | 5300 | 9000 | 4600 |
| 3260 | 40 | 9300 | 26400 | 18000 | 10000 | 115800 | 13000 | 17400 |
| 7345 | 60 | 25300 | 35100 | 37900 | 29400 | 19900 | 17200 | 30600 |
| 11875 | 80 | 38300 | 26400 | 29000 | 33100 | 28200 | 22100 | 36400 |
| 16235 | 100 | 29300 | 25500 | 25300 | 26100 | 35500 | 26000 | 29200 |

TABLE 3a

Specimen III

t = 0.055 With Internal Pressure

| Load lbs | p P.S.I. | x $10^{-3}$ $\delta$" | Gage Millivolts 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|---|
| 10,730 | 50 | 24 | -1.082 | -.292 | -.800 | -1.335 | .530 | .067 | .122 |
| 7,370 | | 34 | - .060 | 1.047 | .491 | - .158 | 1.043 | .962 | .222 |
| 4,100 | | 44 | 1.044 | 2.437 | 1.857 | 1.070 | 1.600 | 1.939 | .292 |
| 14,350 | 100 | 24 | - .738 | .180 | -.248 | - .813 | 1.030 | .712 | N.G. |
| 11,580 | | 34 | .288 | 1.472 | 1.028 | .361 | 1.545 | 1.602 | |
| 9,170 | | 44 | 1.498 | 3.133 | 2.604 | 1.827 | 2.217 | 2.732 | |
| 5,000 | | 54 | 2.482 | 4.420 | 3.818 | 2.956 | 2.642 | 3.540 | |
| 1,900 | | 64 | 3.794 | 6.150 | 5.430 | 4.395 | 3.260 | 4.630 | |
| 20,125 | 150 | 24 | - .228 | .950 | .470 | - .203 | 1.665 | 1.577 | |
| 16,925 | | 34 | .918 | 2.477 | 1.940 | 1.190 | 2.227 | 2.583 | |
| 14,040 | | 44 | 2.000 | 3.928 | 3.325 | 2.500 | 2.773 | 3.542 | |
| 10,525 | | 54 | 3.100 | 5.380 | 4.705 | 3.802 | 3.265 | 4.453 | |
| 7,285 | | 64 | 4.068 | 6.780 | 6.015 | 5.000 | 3.700 | 5.335 | |
| 4,350 | | 74 | 4.932 | 8.240 | 7.353 | 6.042 | 3.867 | 6.218 | |
| 25,575 | 200 | 24 | - .010 | 1.708 | 1.100 | .120 | 1.910 | 2.495 | |
| 22,210 | | 34 | 1.188 | 3.313 | 2.465 | 1.578 | 2.490 | 3.532 | |
| 18,600 | | 44 | 2.322 | 4.320 | 4.070 | 2.957 | 3.022 | 4.463 | |
| 15,320 | | 54 | 3.393 | 6.233 | 5.417 | 4.260 | 3.533 | 5.400 | |
| 17,305 | | 64 | 4.363 | 7.645 | 6.718 | 5.490 | 3.751 | 6.518 | |
| 9,965 | | 74 | 5.190 | 9.793 | 8.265 | 6.718 | 3.821 | 7.610 | |

TABLE 3b

| Load lbs | p P.S.I. | x $10^{-3}$ $\delta$" | Gage Millivolts 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|---|
| 36,880 | 300 | 24 | .681 | 3.947 | 2.755 | 1.250 | 2.845 | 5.034 |
| 33,840 | | 34 | 1.120 | 4.968 | 3.662 | 1.880 | 4.733 | 7.846 |
| 32,200 | | 44 | 2.431 | 6.626 | 5.240 | 3.455 | 5.245 | 8.830 |
| 27,000 | | 54 | 4.138 | 8.712 | 7.320 | 5.525 | 5.210 | 9.561 |
| 25,000 | | 64 | 6.170 | 11.705 | 10.242 | 8.200 | 4.430 | 9.540 |
| 23,320 | | 74 | 8.700 | 15.850 | 14.342 | 11.980 | 3.760 | 9.610 |
| 52,200 | 400 | 24 | 3.690 | 9.546 | 8.374 | 6.030 | 3.377 | 7.534 |
| 48,425 | | 34 | 5.170 | 11.228 | 9.907 | 7.614 | 4.434 | 9.250 |
| 44,750 | | 44 | 6.650 | 12.690 | 11.325 | 9.035 | 4.757 | 10.290 |
| 42,075 | | 54 | 7.680 | 14.680 | 13.570 | 11.076 | 4.764 | 11.260 |
| 40,405 | | 64 | 10.038 | 18.280 | 17.730 | 14.842 | 4.513 | 12.123 |
| 38,580 | | 74 | 13.253 | N.G. | N.G. | 19.200 | 4.414 | 12.572 |

TABLE 3c

Stresses

| Load lbs | p psi | $\delta''$ x $10^{-3}$ | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|---|
| 10730 | 50 | 24 | -12200 | -3200 | -9000 | -15000 | 5900 | 500 |
| 7370 | 50 | 34 | - 400 | 11600 | 5400 | - 1700 | 11600 | 10800 |
| 4100 | 50 | 44 | 11600 | 27500 | 21200 | 12100 | 18000 | 21900 |
| 14350 | 100 | 24 | - 8200 | 1800 | -2700 | - 9100 | 11500 | 8000 |
| 11580 | 100 | 34 | 3200 | 16600 | 11400 | 4000 | 17400 | 18000 |
| 9170 | 100 | 44 | 16900 | 35400 | 29400 | 20500 | 25000 | 31000 |
| 5000 | 100 | 54 | 28100 | 33000 | 37800 | 33500 | 29800 | 38300 |
| 1900 | 100 | 64 | 37900 | 26100 | 28000 | 33200 | 37000 | 31700 |
| 20125 | 150 | 24 | - 2500 | 10700 | 5200 | - 2100 | 18600 | 17700 |
| 16925 | 150 | 34 | 10200 | 28100 | 22000 | 13300 | 25200 | 29200 |
| 14040 | 150 | 44 | 22700 | 37000 | 37700 | 28300 | 31400 | 38300 |
| 10525 | 150 | 54 | 35100 | 28100 | 31100 | 37800 | 37100 | 32800 |
| 7285 | 150 | 64 | 36000 | 25400 | 26400 | 29700 | 38200 | 28300 |
| 4350 | 150 | 74 | 30000 | 24800 | 25300 | 26300 | 37400 | 26000 |
| 25575 | 200 | 24 | -- | 19200 | 12400 | 1100 | 21600 | 28100 |
| 22210 | 200 | 34 | 13300 | 37400 | 27800 | 12200 | 28100 | 38300 |
| 18600 | 200 | 44 | 26200 | 30700 | 35900 | 33600 | 34100 | 32700 |
| 15320 | 200 | 54 | 38200 | 26000 | 28000 | 34200 | 38300 | 28100 |
| 12305 | 200 | 64 | 33400 | 25400 | 25400 | 27700 | 38000 | 25600 |
| 9965 | 200 | 74 | 28800 | 27800 | 25800 | 25400 | 37700 | 25300 |
| 36880 | 300 | 24 | 7600 | 36800 | 31200 | 14000 | 32100 | 29600 |
| 33840 | 300 | 34 | 12500 | 29900 | 38200 | 21300 | 30900 | 25500 |
| 30220 | 300 | 44 | 27500 | 25500 | 28600 | 38300 | 38600 | 26400 |
| 27000 | 300 | 54 | 35200 | 26200 | 25300 | 27600 | 28700 | 27500 |
| 25000 | 300 | 64 | 28100 | 31800 | 28600 | 25700 | 33000 | 27400 |
| 23320 | 200 | 74 | 26100 | 39400 | 36800 | 32300 | 38000 | 27600 |

TABLE 3d

Stresses

| Load lbs | p p.s.i. | $\delta''$ $\times 10^{-3}$ | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|---|---|
| 52200 | 400 | 24 | 38200 | 26700 | 25900 | 26400 | 38000 | 25300 |
| 48425 | 400 | 34 | 28800 | 30700 | 28000 | 25300 | 32900 | 27000 |
| 44750 | 400 | 44 | 25400 | 33700 | 30900 | 26600 | 30900 | 28800 |
| 42075 | 400 | 54 | 25400 | 37400 | 35400 | 30300 | 30800 | 30800 |
| 40405 | 400 | 64 | 28200 | 43700 | 42100 | 37700 | 32500 | 32600 |
| 38580 | 400 | 74 | 34800 | | | 44500 | 35300 | 33400 |

TABLE 4a

Specimen IV

t = 0.055 With Internal Pressure

| Load lbs | psi p | x $10^{-3}$ δ" | Gage Millivolts | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 1 | 2 | 3 | 4 | 6 | 7 | 8 | 9 | 10 |
| 8270 | 50 | 0 | .278 | .465 | .550 | .412 | .885 | 1.072 | .622 | -.328 | -.178 |
| 6355 | | 10 | 1.374 | 1.540 | 1.603 | 1.513 | 1.281 | 1.860 | .965 | -.392 | -.047 |
| 4150 | | 20 | 2.142 | 2.412 | 2.496 | 2.417 | 1.568 | 2.502 | 1.186 | -.415 | +.082 |
| 2500 | | 30 | 2.923 | 3.277 | 3.360 | 3.324 | 1.824 | 3.115 | 1.422 | -.455 | +.200 |
| 1650 | | 35 | 3.420 | 3.852 | 3.988 | 4.096 | 1.773 | 3.390 | 1.508 | -.462 | +.277 |
| 13000 | 100 | 0 | 1.575 | 1.070 | 1.518 | 1.510 | 1.000 | 1.273 | .850 | -.512 | -.263 |
| 11800 | | 10 | 2.146 | 2.082 | 2.400 | 2.429 | 1.448 | 2.210 | 1.231 | -.575 | -.136 |
| 8675 | | 20 | 2.735 | 2.983 | 3.200 | 3.259 | 1.742 | 2.933 | 1.465 | -.581 | +.030 |
| 6500 | | 30 | 3.313 | 3.905 | 3.817 | 4.088 | 2.067 | 3.702 | 1.741 | -.604 | +.154 |
| 5270 | | 40 | 3.792 | 4.700 | 4.828 | 4.935 | 2.093 | 4.361 | 1.960 | -.653 | +.274 |
| 3650 | | 50 | 4.118 | 5.377 | 5.482 | 5.746 | 2.130 | 4.800 | 2.073 | -.662 | +.402 |
| 2715 | | 60 | 4.320 | 6.077 | 6.073 | 6.642 | 2.322 | 5.250 | 2.220 | -.679 | .555 |
| 1500 | | 70 | 4.508 | 6.700 | 6.558 | 7.431 | 2.688 | 5.363 | 2.245 | -.687 | .727 |

## TABLE 4b

Specimen IV

| Load lbs | P psi | $\delta''$ $\times 10^{-3}$ | Gage Millivolts 1 | 2 | 3 | 4 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 25,100 | 200 | 10 | 1.448 | 2.390 | 2.797 | 3.440 | 1.055 | 2.753 | 1.752 | - 1.060 | - .432 |
| 23,400 | | 20 | 2.075 | 3.400 | 3.698 | 4.353 | 1.457 | 3.547 | 2.045 | - 1.122 | - .310 |
| 20,625 | | 30 | 2.682 | 4.337 | 4.530 | 5.219 | 1.720 | 4.239 | 2.290 | - 1.145 | - .163 |
| 19,180 | | 40 | 3.137 | 5.080 | 5.177 | 5.950 | 1.942 | 5.024 | 2.481 | - 1.210 | - .058 |
| 17,670 | | 50 | 3.572 | 5.832 | 5.962 | 6.640 | 1.937 | 5.618 | 2.470 | - 1.280 | 000 |
| 16,200 | | 60 | 4.110 | 6.600 | 6.917 | 7.440 | 1.720 | 6.135 | 2.328 | - 1.450 | + .015 |
| 15,750 | | 70 | 4.672 | 7.500 | 7.988 | 8.480 | 1.413 | 6.362 | 2.045 | - 1.565 | + .070 |
| | | | | | | | | | | | |
| 41,000 | 300 | 10 | .842 | 2.292 | 2.950 | 3.452 | .381 | 2.756 | 1.218 | - 1.920 | -1.170 |
| 38,670 | | 20 | 1.468 | 3.462 | 3.970 | 4.572 | .917 | 3.631 | 1.612 | - 2.010 | -1.046 |
| 35,170 | | 30 | 2.155 | 4.475 | 4.988 | 5.591 | 1.328 | 4.508 | 1.920 | - 2.050 | - .892 |
| 32,950 | | 40 | 2.768 | 5.457 | 5.868 | 6.495 | 1.830 | 5.527 | 2.228 | - 2.150 | - .832 |
| 31,600 | | 50 | 3.175 | 6.141 | 6.661 | 7.233 | 1.946 | 6.142 | 2.254 | - 2.270 | - .812 |
| 30,850 | | 60 | 3.812 | 7.233 | 7.730 | 8.188 | 2.013 | 6.940 | 1.955 | - 2.578 | - .945 |
| | | | | | | | | | | | |
| 67,500 | 500 | 10 | -2.928 | 3.238 | 2.724 | 3.090 | 7.725 | 7.727 | 6.020 | - 8.327 | -7.012 |
| 64,500 | | 20 | -2.225 | 4.963 | 4.249 | 4.940 | 8.757 | 8.736 | 5.970 | - 9.308 | -7.330 |
| 63,875 | | 30 | -1.639 | 6.917 | 5.622 | 6.912 | 9.220 | 8.500 | 5.902 | -10.383 | -7.910 |
| 62,760 | | 40 | - .850 | 9.296 | 7.449 | 9.420 | 9.400 | 7.250 | 5.579 | -11.260 | -8.221 |
| 60,500 | | 50 | + .294 | 12.200 | 9.762 | 12.217 | 9.632 | 5.457 | 5.326 | -12.348 | -8.698 |
| 58,135 | | 60 | 1.500 | | | 15.283 | 9.345 | 4.727 | 4.850 | -12.819 | -8.725 |
| 56,200 | | 70 | 3.398 | | | 19.250 | 9.000 | 3.639 | 4.310 | -13.215 | -8.773 |
| 55,305 | | 80 | 5.360 | | | | 8.728 | 2.255 | 3.810 | -13.752 | -8.812 |
| 53,870 | | 90 | 8.540 | | | | 8.378 | 1.500 | 3.333 | -14.072 | -8.770 |
| 52,000 | | 100 | | | | | 7.995 | .903 | 2.896 | -14.257 | -8.609 |

TABLE 4c

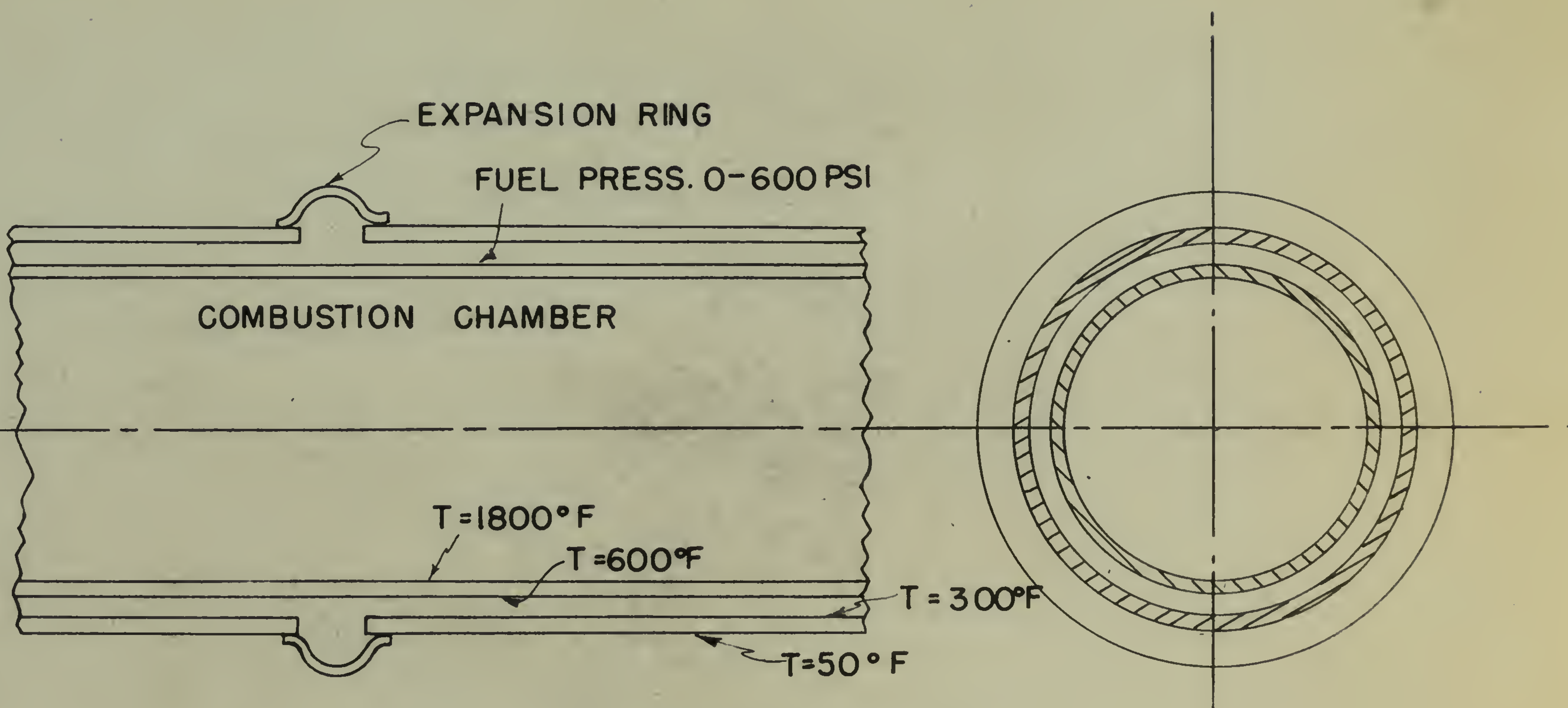
Specimen IV

| Load lbs | p psi | $\delta''$ x $10^{-3}$ | Gage Stresses 1 | 2 | 3 | 4 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 8270 | 50 | 0 | 3000 | 5000 | 6100 | 4600 | 9900 | 12100 | 7000 | -3500 | -1800 |
| 6355 | | 10 | 15500 | 17400 | 18000 | 17000 | 14400 | 21000 | 10800 | -4200 | - 200 |
| 4150 | | 20 | 24200 | 27300 | 28200 | 27300 | 17700 | 28200 | 13200 | -4500 | 700 |
| 2500 | | 30 | 33000 | 37100 | 38100 | 37600 | 20400 | 35200 | 16000 | -5000 | 2100 |
| 1650 | | 35 | 38200 | 37600 | 36500 | 35700 | 19900 | 38200 | 17000 | -5200 | 3000 |
| 13000 | 100 | 0 | 17800 | 12100 | 17100 | 17000 | 11200 | 14100 | 9500 | -5700 | -2800 |
| 11800 | | 10 | 24000 | 23500 | 27200 | 27500 | 16300 | 25000 | 13800 | -6400 | -1300 |
| 8675 | | 20 | 30800 | 33900 | 36200 | 37000 | 19600 | 33100 | 16400 | -6500 | 100 |
| 6500 | | 30 | 37400 | 37100 | 37800 | 35700 | 23300 | 38200 | 19700 | -6600 | 1600 |
| 5270 | | 40 | 37800 | 31200 | 30500 | 30000 | 23600 | 33400 | 22200 | -7100 | 3000 |
| 3650 | | 50 | 35500 | 28100 | 27800 | 27000 | 24100 | 30600 | 23500 | -7200 | 4400 |
| 2715 | | 60 | 33800 | 26200 | 26300 | 25500 | 26100 | 28600 | 25000 | -7500 | 6100 |
| 1500 | | 70 | 32400 | 25400 | 25600 | 25300 | 30300 | 28200 | 25300 | -7600 | 8000 |

TABLE 4d

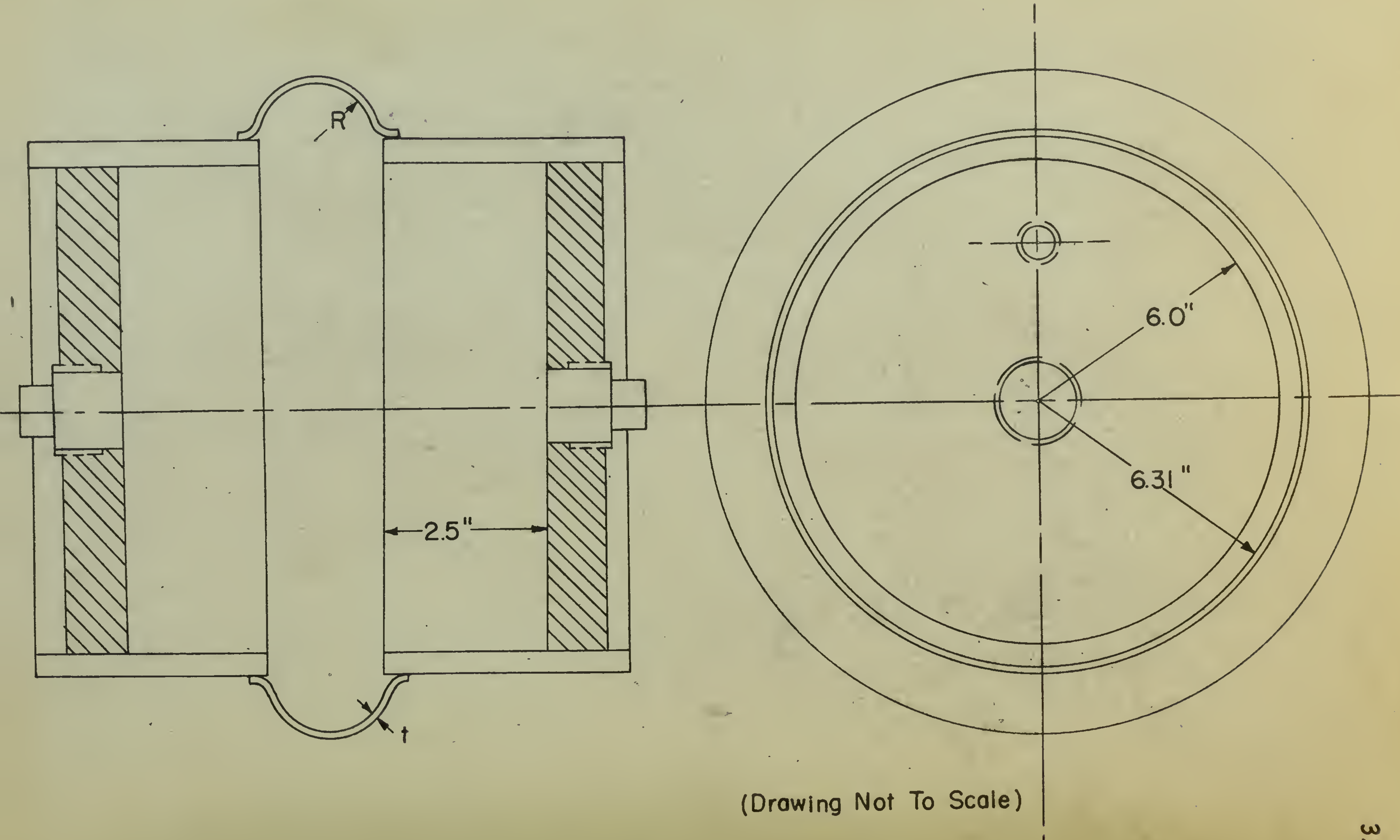
Specimen IV

| Load lbs | p p.s.i. | x 10^-3 δ" | Gage Stresses 1 | 2 | 3 | 4 | 6 | 7 | 8 | 9 | 10 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 25100 | 200 | 10 | 16400 | 27100 | 31600 | 38300 | 11600 | 31200 | 19800 | -11900 | - 4700 |
| 23400 | | 20 | 23400 | 38200 | 38200 | 33400 | 16400 | 38300 | 23000 | -12600 | - 3400 |
| 20625 | | 30 | 30300 | 33600 | 32400 | 28700 | 19500 | 34400 | 26000 | -12900 | - 1600 |
| 19180 | | 40 | 35400 | 29400 | 28900 | 26500 | 22000 | 29600 | 28100 | -13400 | - 100 |
| 17670 | | 50 | 38300 | 26800 | 26500 | 25500 | 21900 | 27400 | 28000 | -14400 | 0 |
| 16200 | | 60 | 35500 | 25500 | 25300 | 25300 | 19500 | 26200 | 26200 | -16500 | 100 |
| 15750 | | 70 | 31400 | 25300 | 25600 | 26000 | 15900 | 25800 | 23000 | -17700 | 200 |
| 41000 | 300 | 10 | 9400 | 25900 | 33400 | 38300 | 4200 | 31200 | 13700 | -21600 | -13200 |
| 38670 | | 20 | 16500 | 38300 | 36700 | 32000 | 10200 | 38300 | 18100 | -22700 | -11500 |
| 35170 | | 30 | 24500 | 32600 | 29700 | 27400 | 14900 | 32500 | 21600 | -23200 | - 9900 |
| 32950 | | 40 | 31200 | 27900 | 26700 | 25600 | 20600 | 27700 | 25200 | -24300 | - 9400 |
| 31600 | | 50 | 35900 | 26200 | 25500 | 25300 | 22100 | 26100 | 25600 | -25600 | - 9200 |
| 30850 | | 60 | 37800 | 25300 | 25400 | 25700 | 22800 | 25300 | 22200 | -29200 | -10700 |
| 67500 | 500 | 10 | -33000 | 36500 | 30800 | 35000 | 25400 | 25400 | 26400 | -25900 | -25300 |
| 64500 | | 20 | -25200 | 29800 | 34300 | 29900 | 26300 | 26200 | 26500 | -27000 | -25300 |
| 63875 | | 30 | -18400 | 25300 | 27400 | 25300 | 26900 | 26000 | 26600 | -28900 | -25500 |
| 62760 | | 40 | - 9500 | 27000 | 25300 | 27200 | 27200 | 25300 | 27500 | -30800 | -25700 |
| 60500 | | 50 | 3200 | 32700 | 22800 | 32700 | 27600 | 27900 | 28400 | -33000 | -26200 |
| 58135 | | 60 | 11200 | | | 38400 | 27100 | 31000 | 30400 | -33900 | -26250 |
| 56200 | | 70 | 38200 | | | 44500 | 26600 | 38300 | 33900 | -34700 | -26300 |
| 55305 | | 80 | 28200 | | | | 26200 | 25600 | 37800 | -35700 | -26400 |
| 53870 | | 90 | 26000 | | | | 25900 | 16900 | 37200 | -36300 | -26300 |
| 52000 | | 100 | | | | | 25500 | 10200 | 32700 | -36600 | -26100 |




ROCKET SECTION
WITH
ASSUMED TEMPERATURE
DISTRIBUTION

FIG. a




(Drawing Not To Scale)

FIG. b

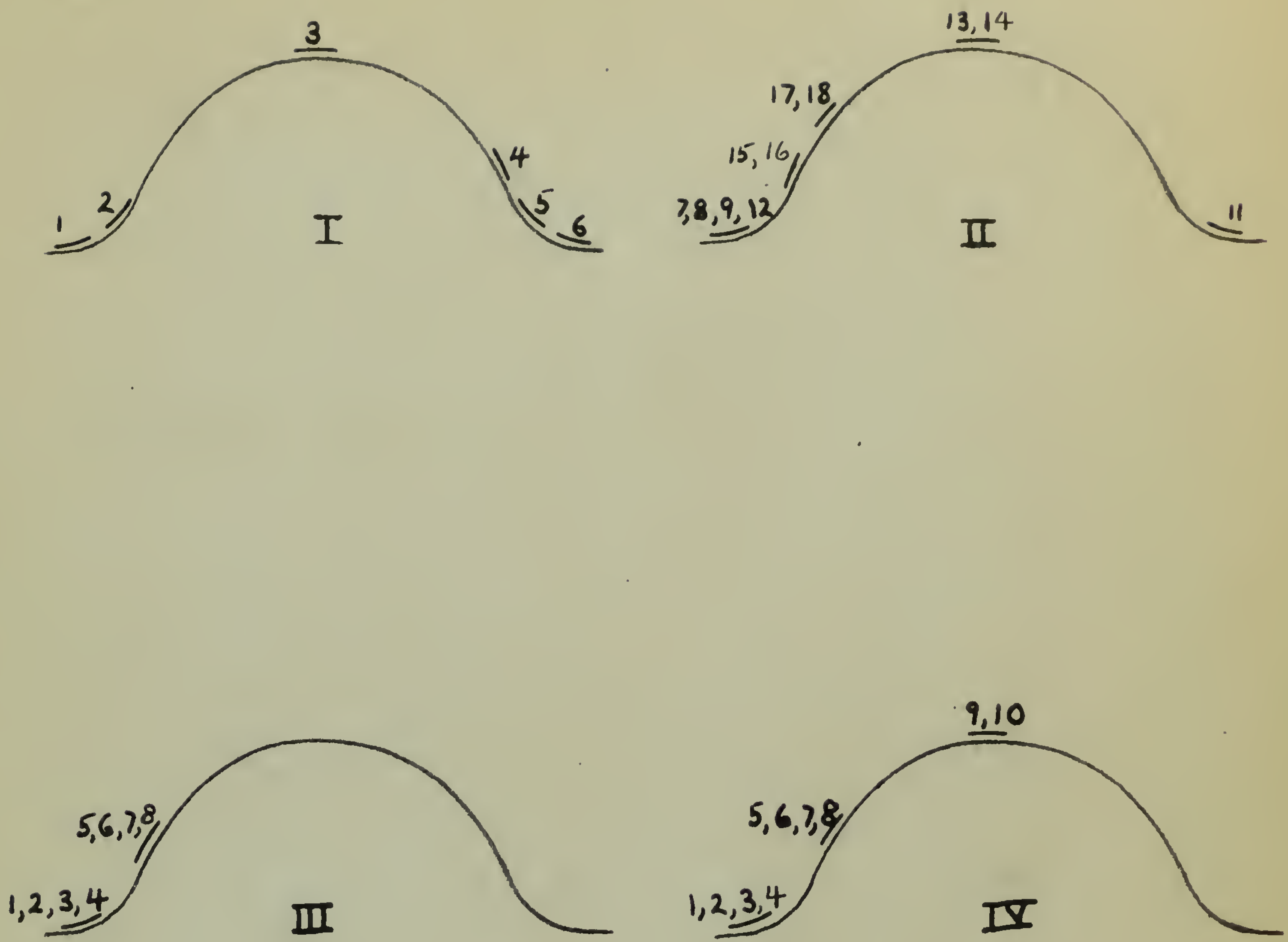
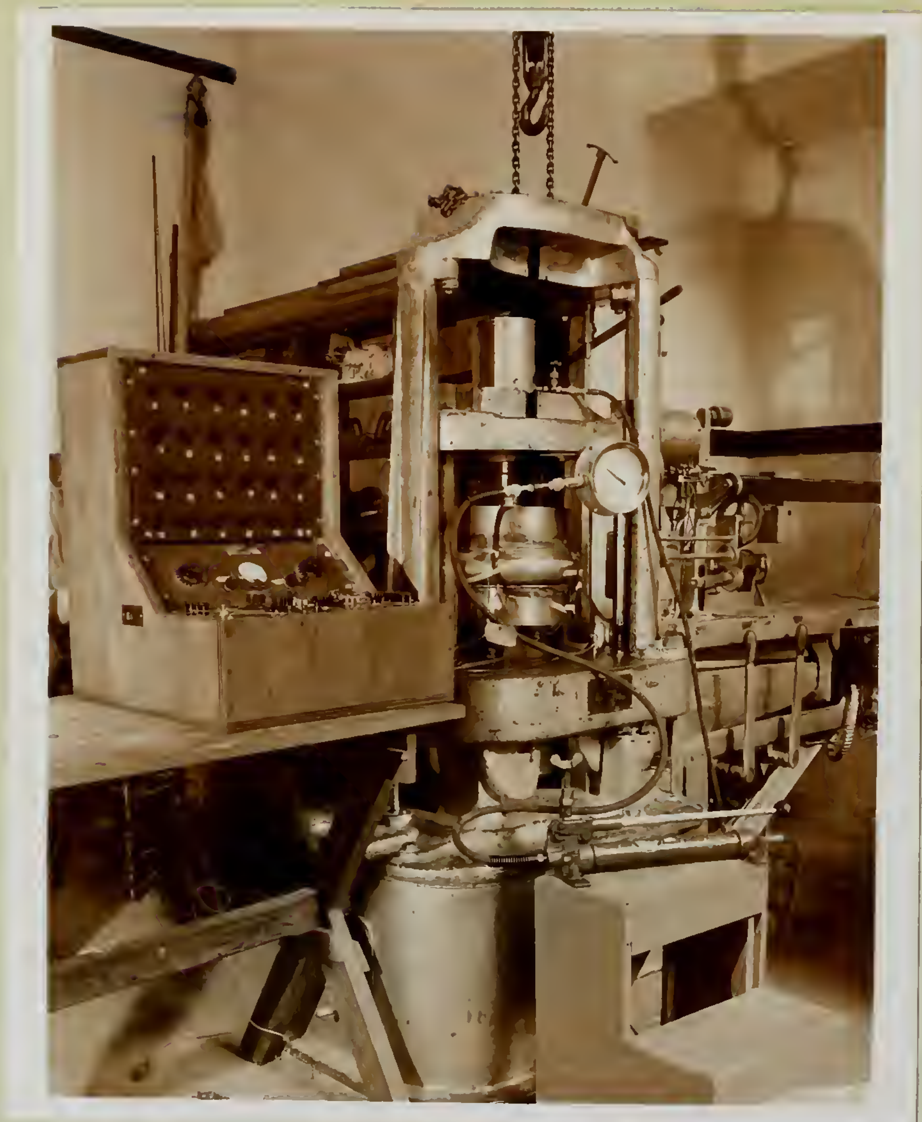
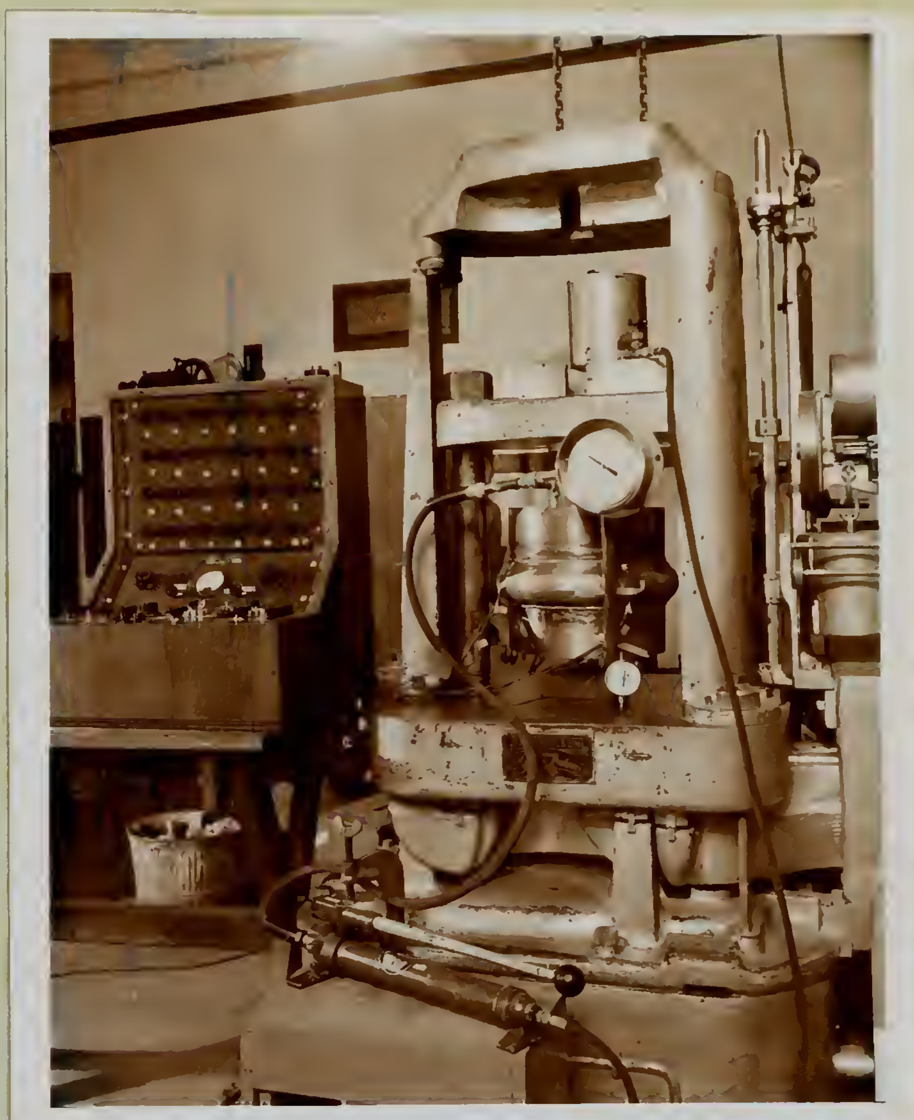

3
1
2
4
5
6
I
13, 14
17, 18
15, 16
7, 8, 9, 12
11
II
5, 6, 7, 8
1, 2, 3, 4
III
9, 10
5, 6, 7, 8
1, 2, 3, 4
IV

Figure C.



Photograph I



Photograph II

Photograph III

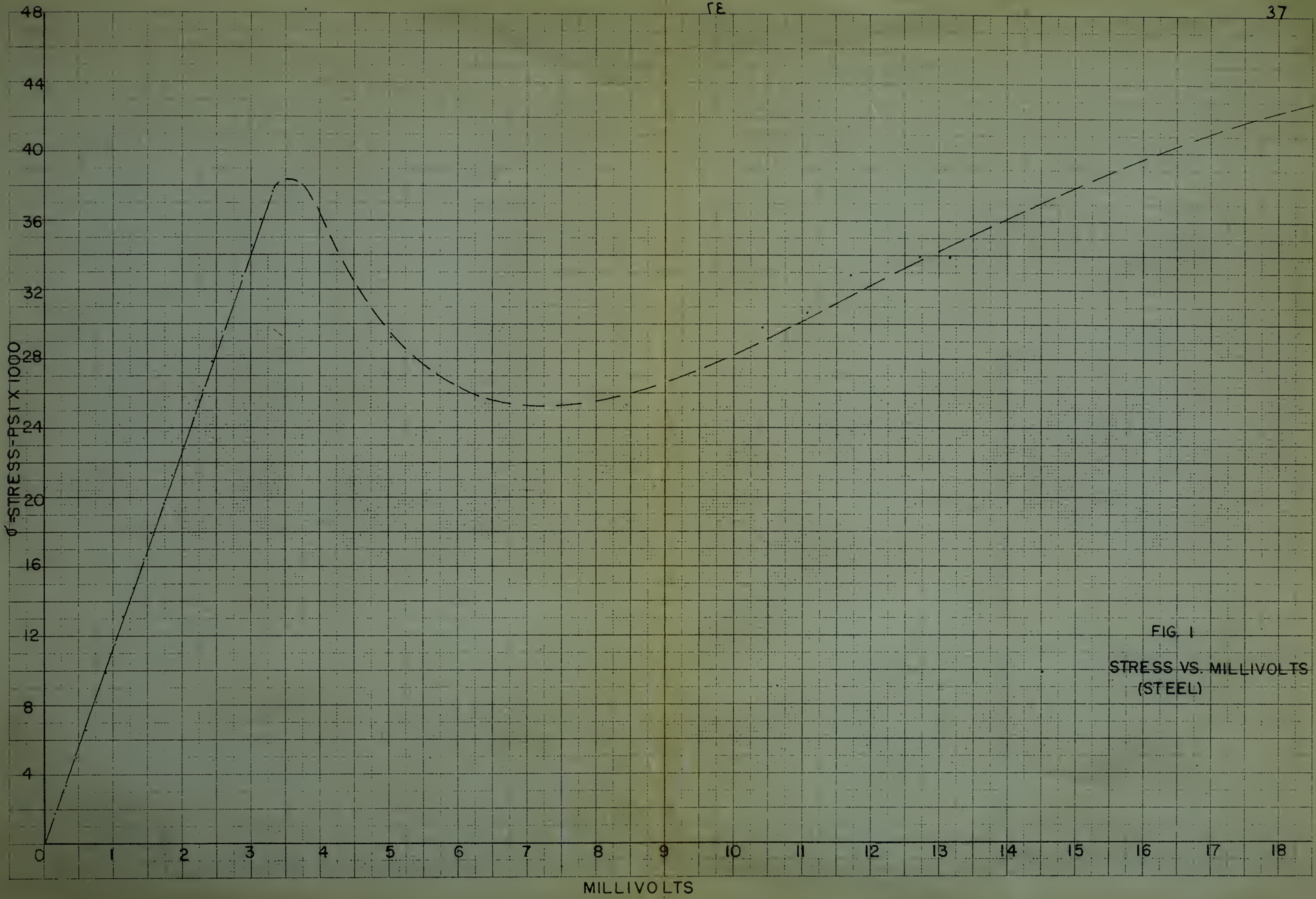



FIG. 1

STRESS VS. MILLIVOLTS
(STEEL)

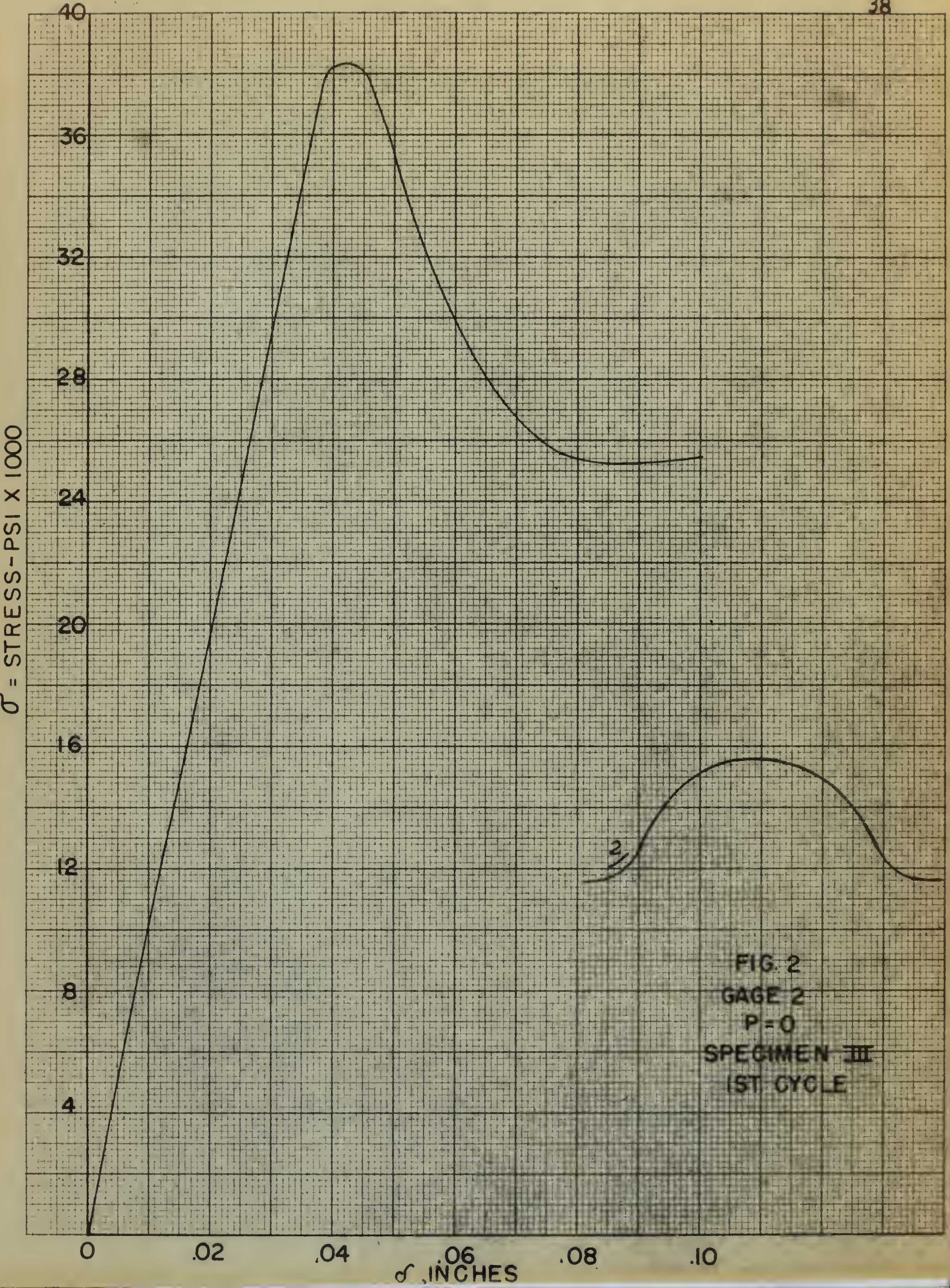



FIG. 2
GAGE 2
P = 0
SPECIMEN III
1ST CYCLE

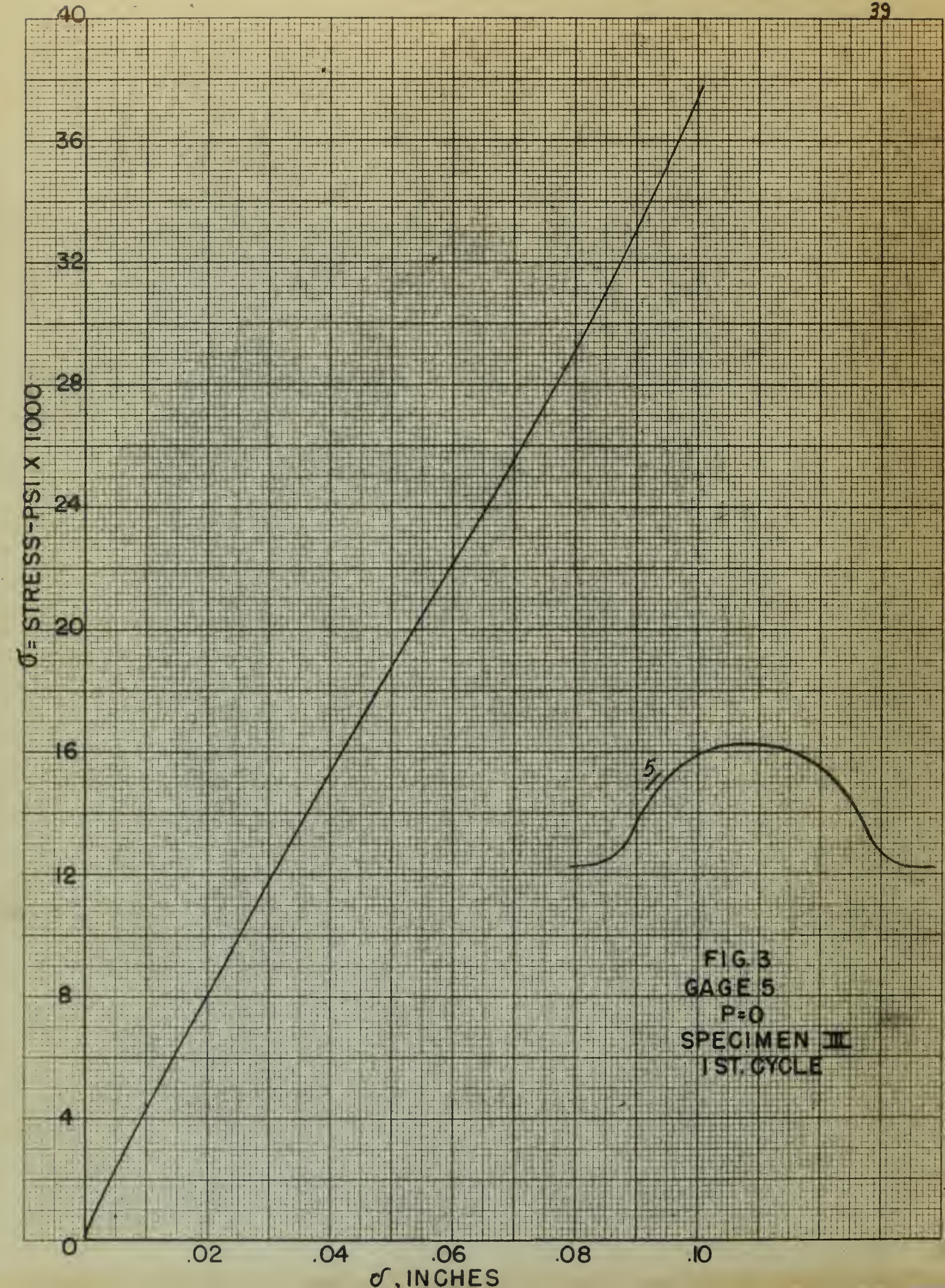
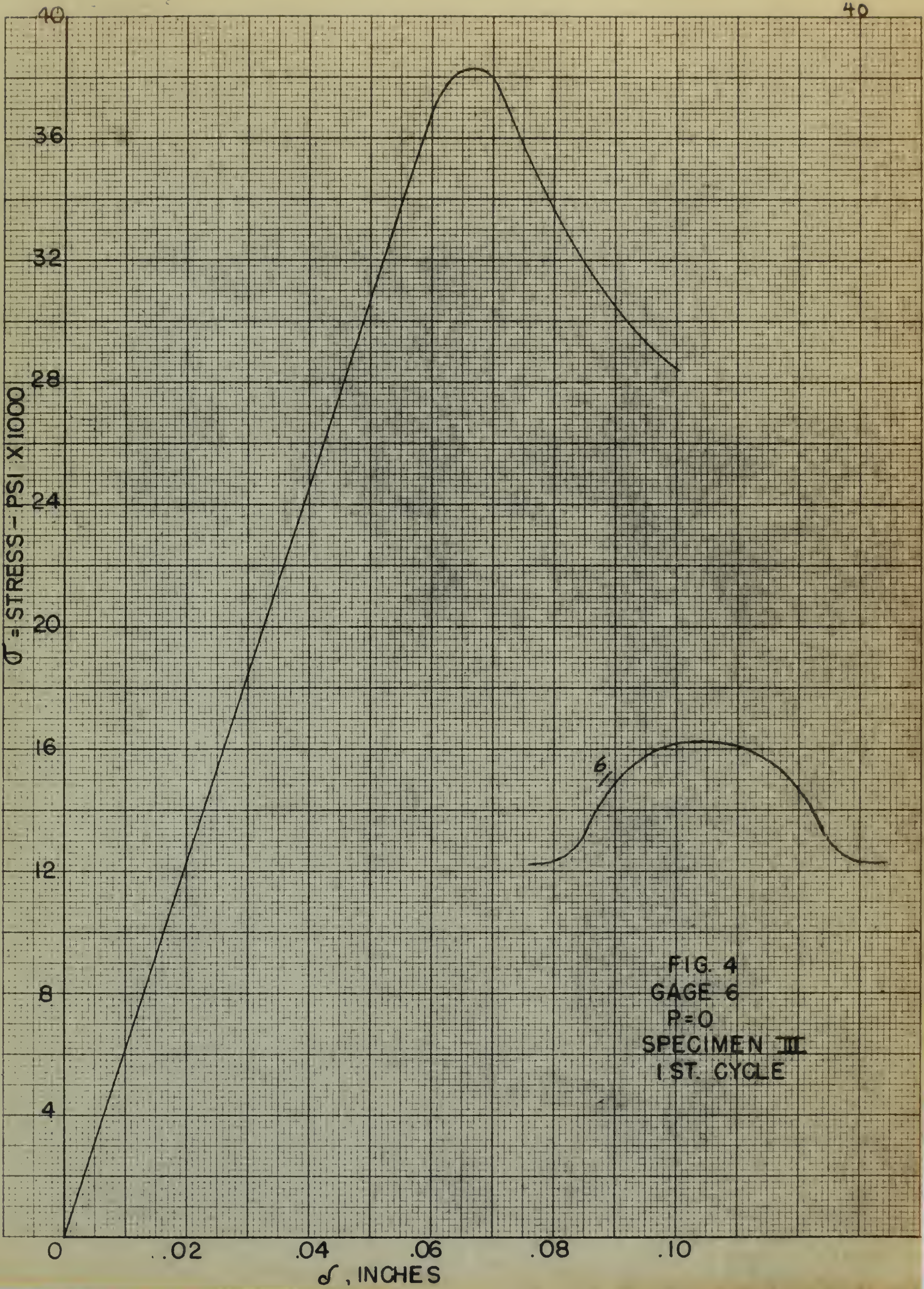



FIG. 3
GAGE 5
P=0
SPECIMEN III
1 ST. CYCLE



40
40
36
32
28
24
20
16
12
8
4
0
.02
.04
.06
.08
.10
σ = STRESS - PSI X 1000
δ, INCHES
6
FIG. 4
GAGE 6
P=0
SPECIMEN III
1 ST. CYCLE

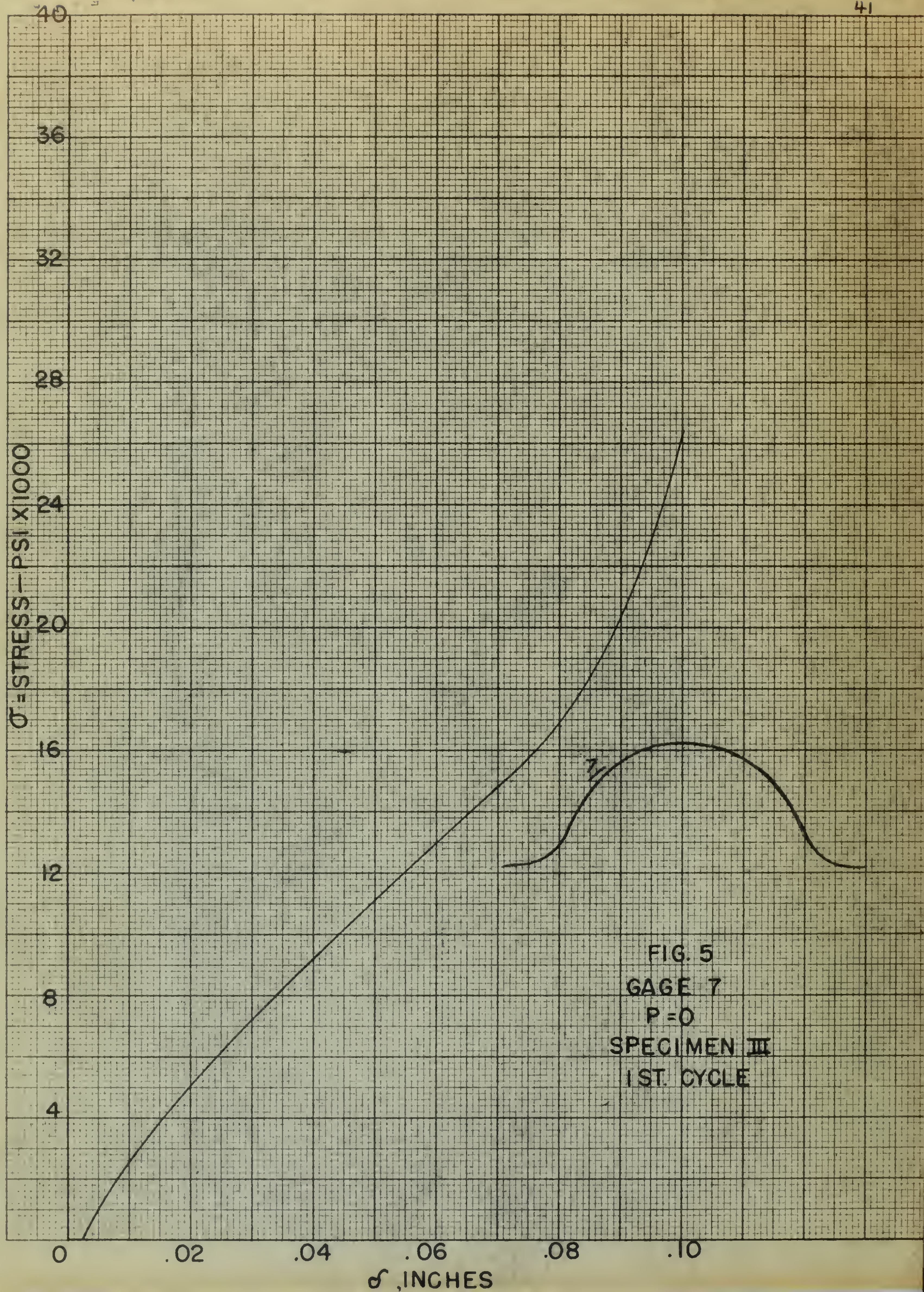
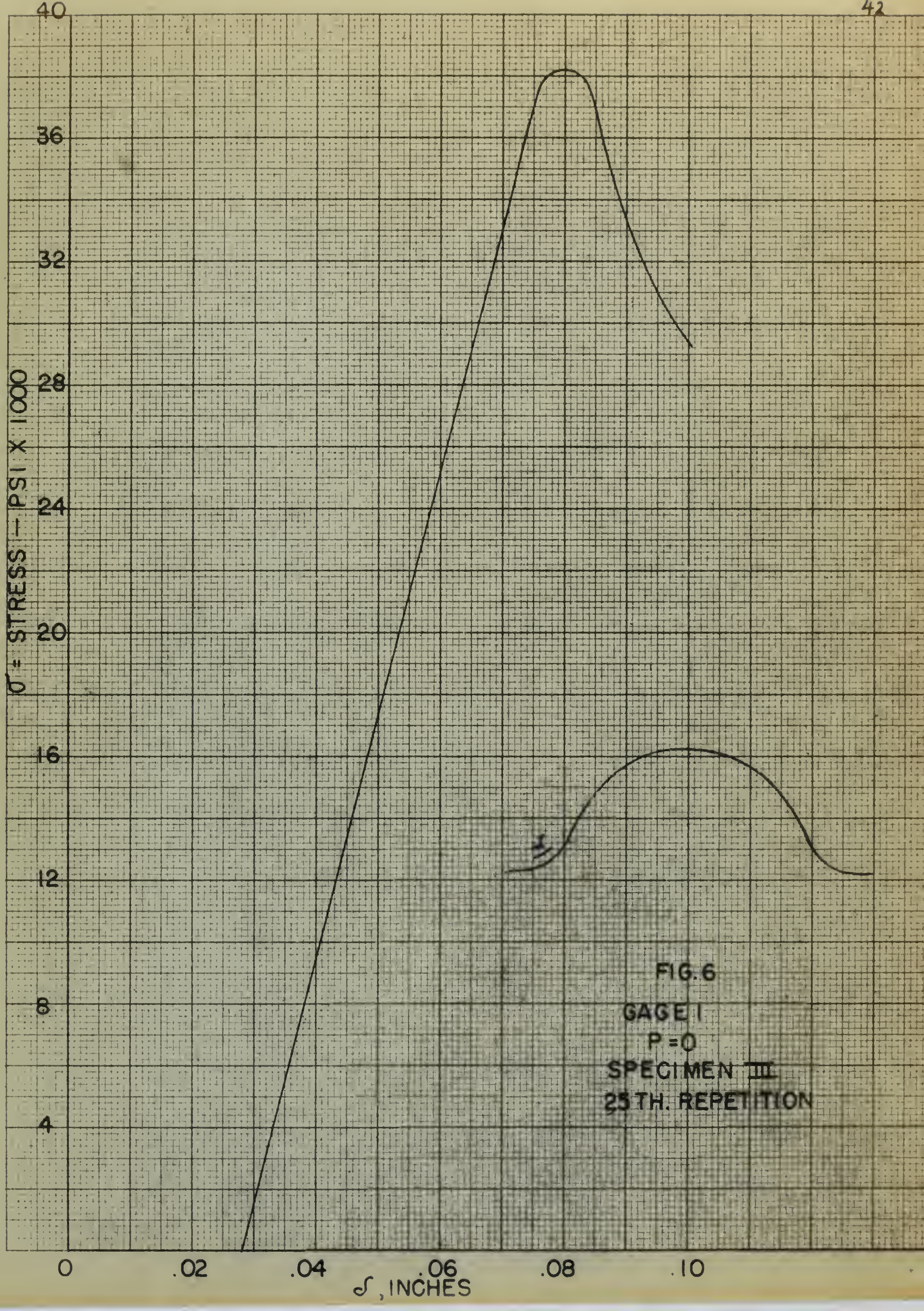
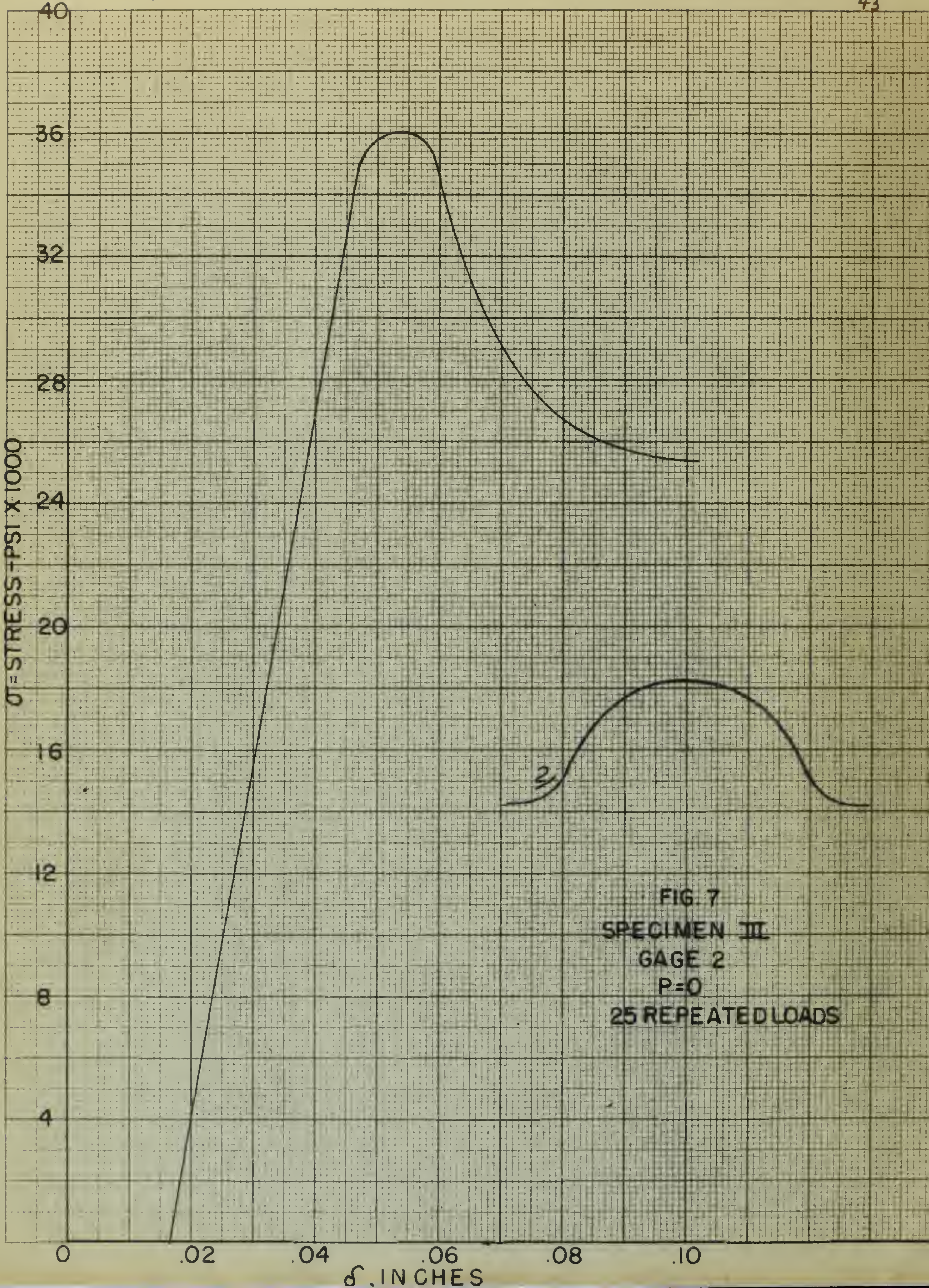



FIG. 5
GAGE 7
P = 0
SPECIMEN III
1 ST. CYCLE


40
36
32
28
24
20
16
12
8
4
0
.02
.04
.06
.08
.10
σ = STRESS — PSI X 1000
δ, INCHES
FIG. 6
GAGE I
P = 0
SPECIMEN III
25TH. REPETITION



40
36
32
28
24
20
16
12
8
4
0
.02
.04
.06
.08
.10
σ = STRESS - PSI X 1000
δ, INCHES
2
FIG. 7
SPECIMEN III
GAGE 2
P=0
25 REPEATED LOADS

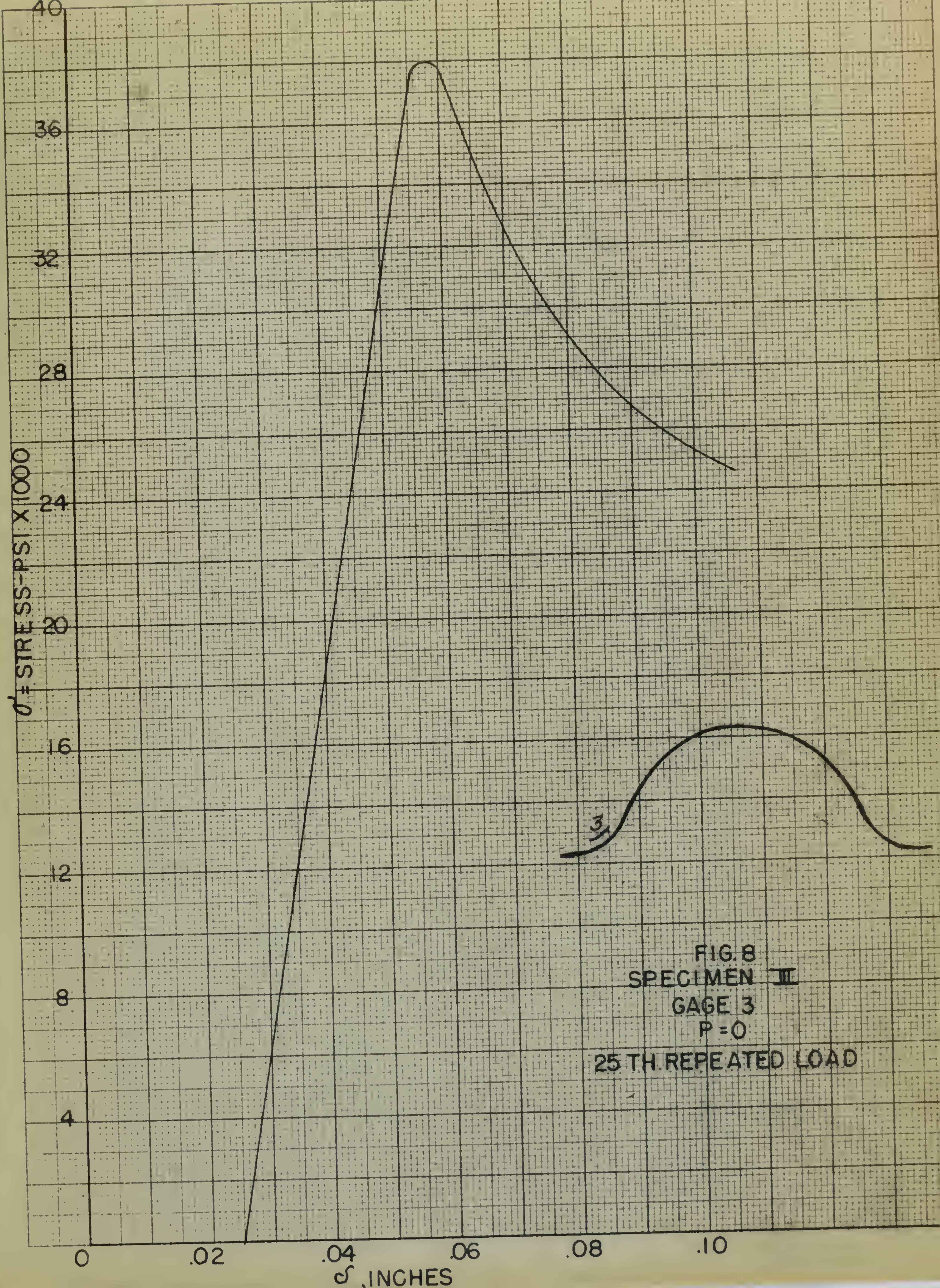



FIG. 8
SPECIMEN III
GAGE 3
P = 0
25 TH. REPEATED LOAD

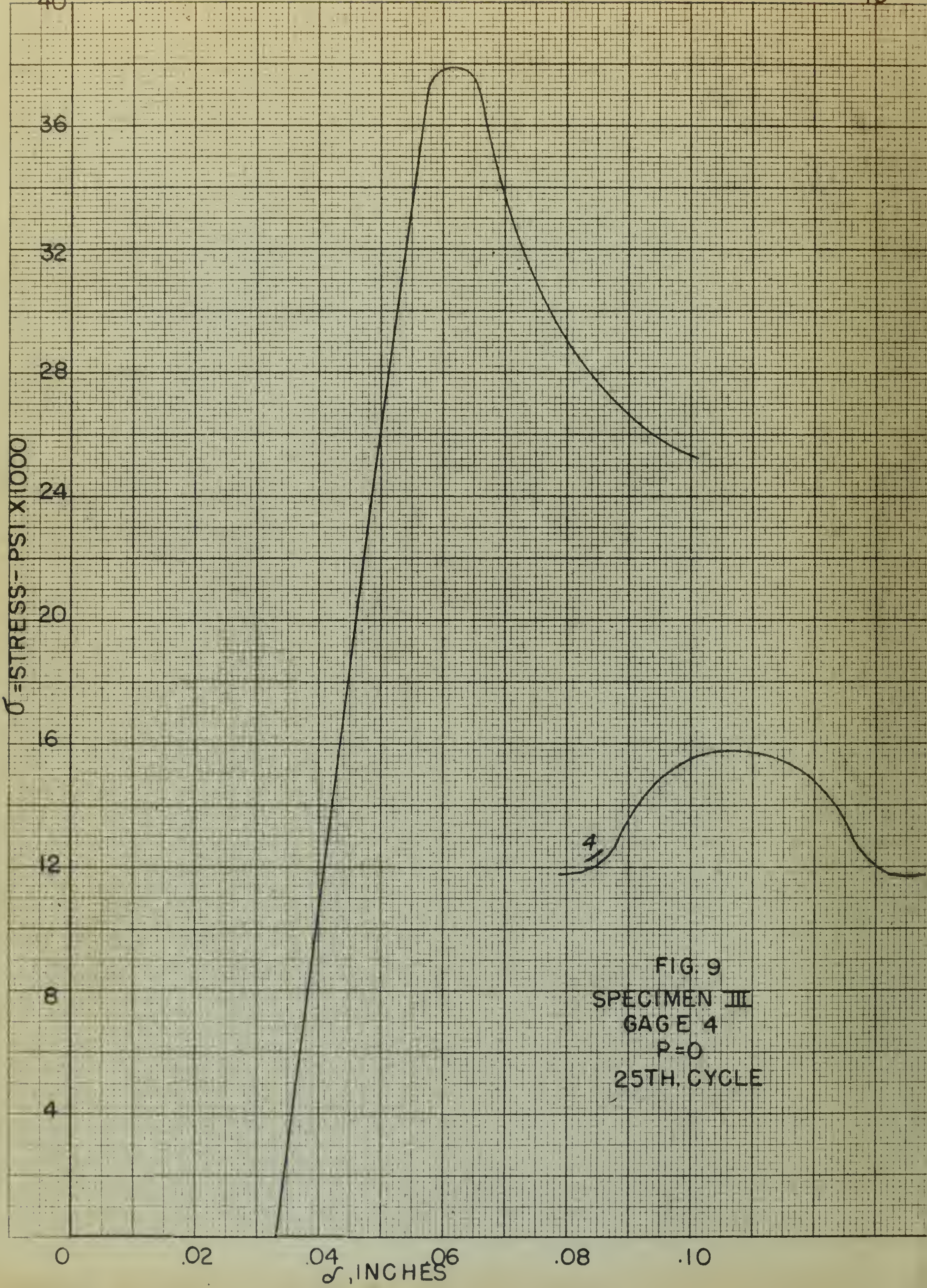
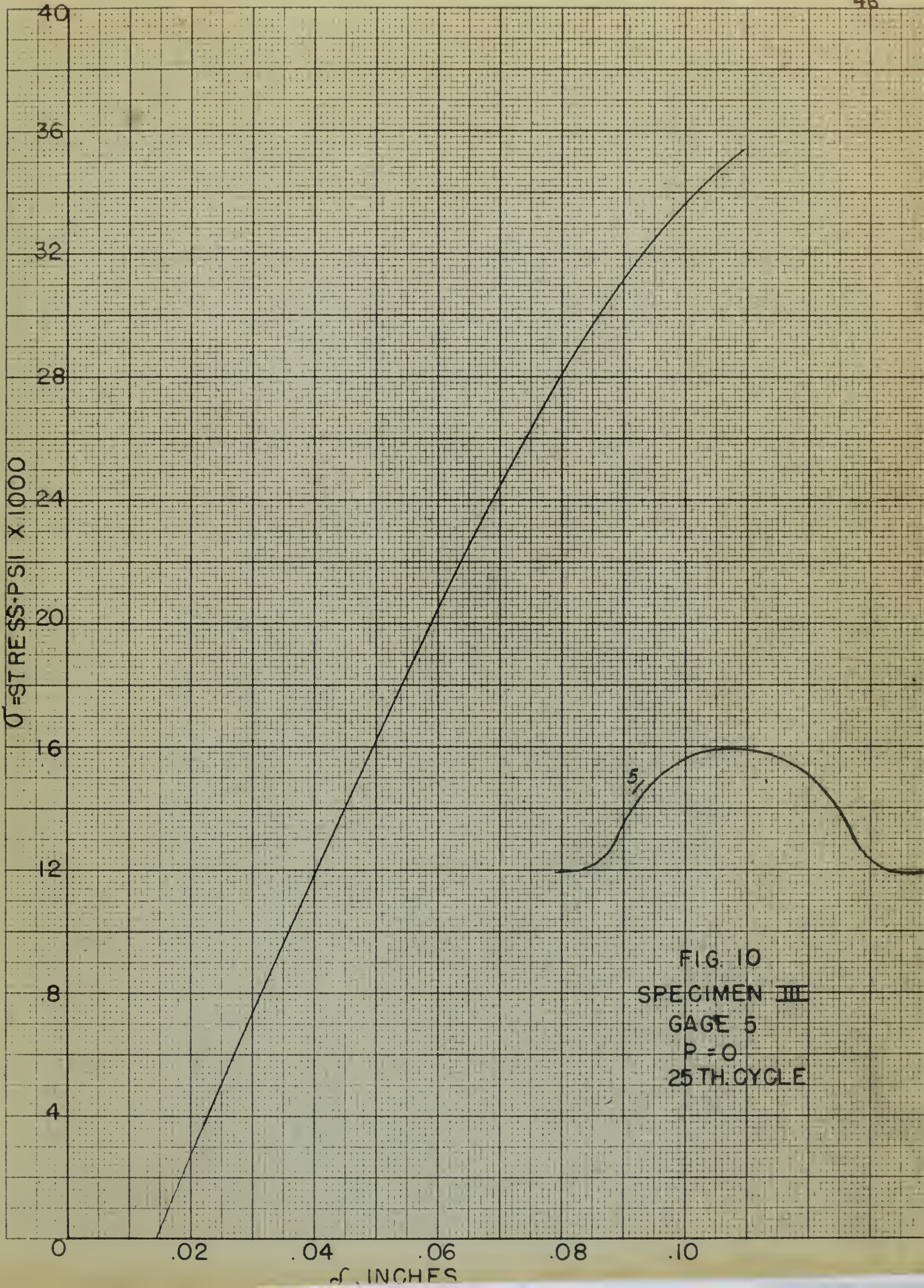
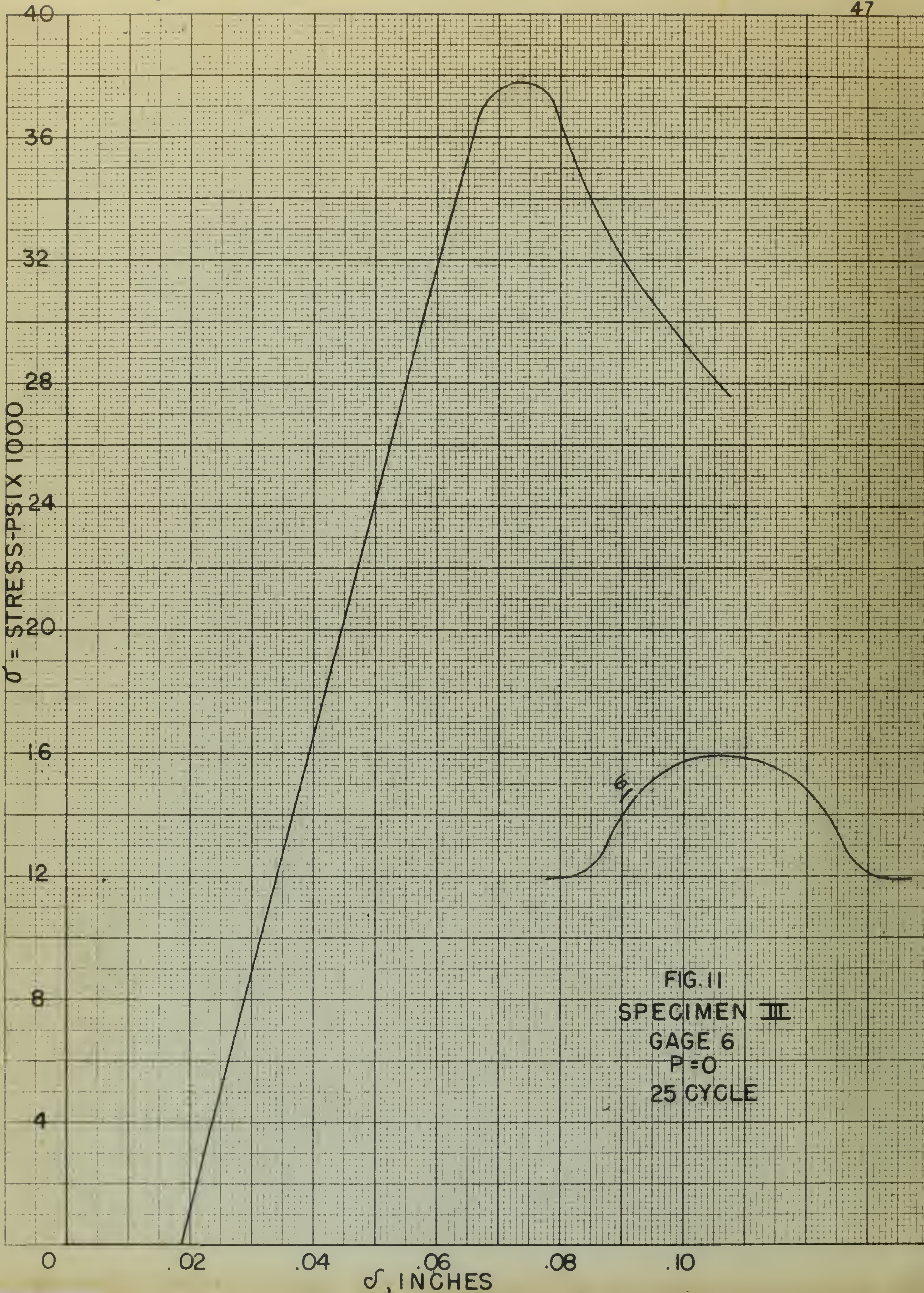



FIG. 9
SPECIMEN III
GAGE 4
P=0
25TH. CYCLE



σ = STRESS-PSI X 1000
40
36
32
28
24
20
16
12
8
4
0
.02
.04
.06
.08
.10
δ, INCHES
5
FIG. 10
SPECIMEN III
GAGE 5
P = 0
25 TH. CYCLE


40
36
32
28
24
20
16
12
8
4
σ = STRESS-PSI X 1000
0
.02
.04
.06
.08
.10
δ, INCHES
61
FIG. 11
SPECIMEN III
GAGE 6
P = 0
25 CYCLE

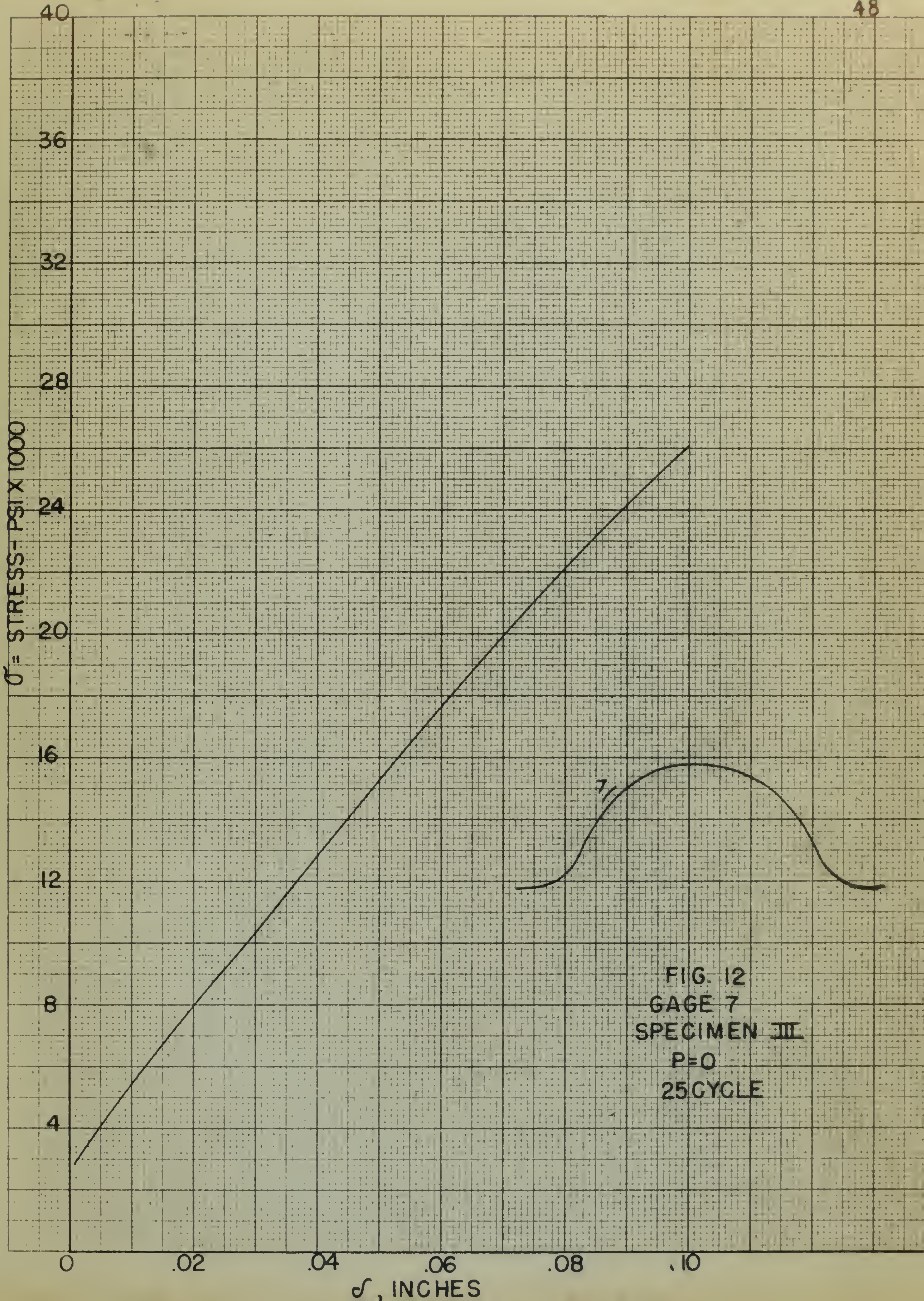



FIG. 12
GAGE 7
SPECIMEN III
P=0
25 CYCLE

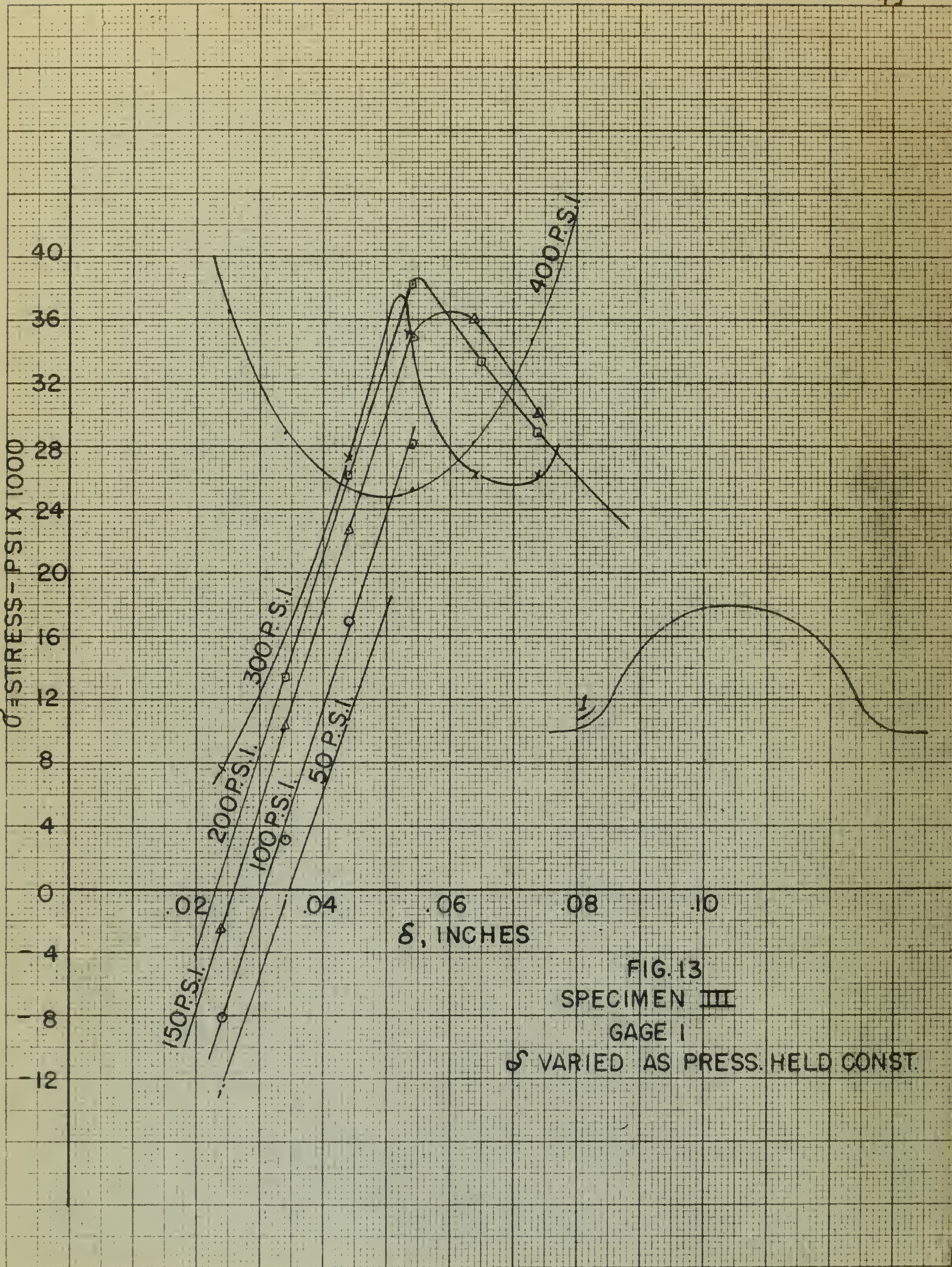



FIG. 13
SPECIMEN III
GAGE I
δ VARIED AS PRESS. HELD CONST.

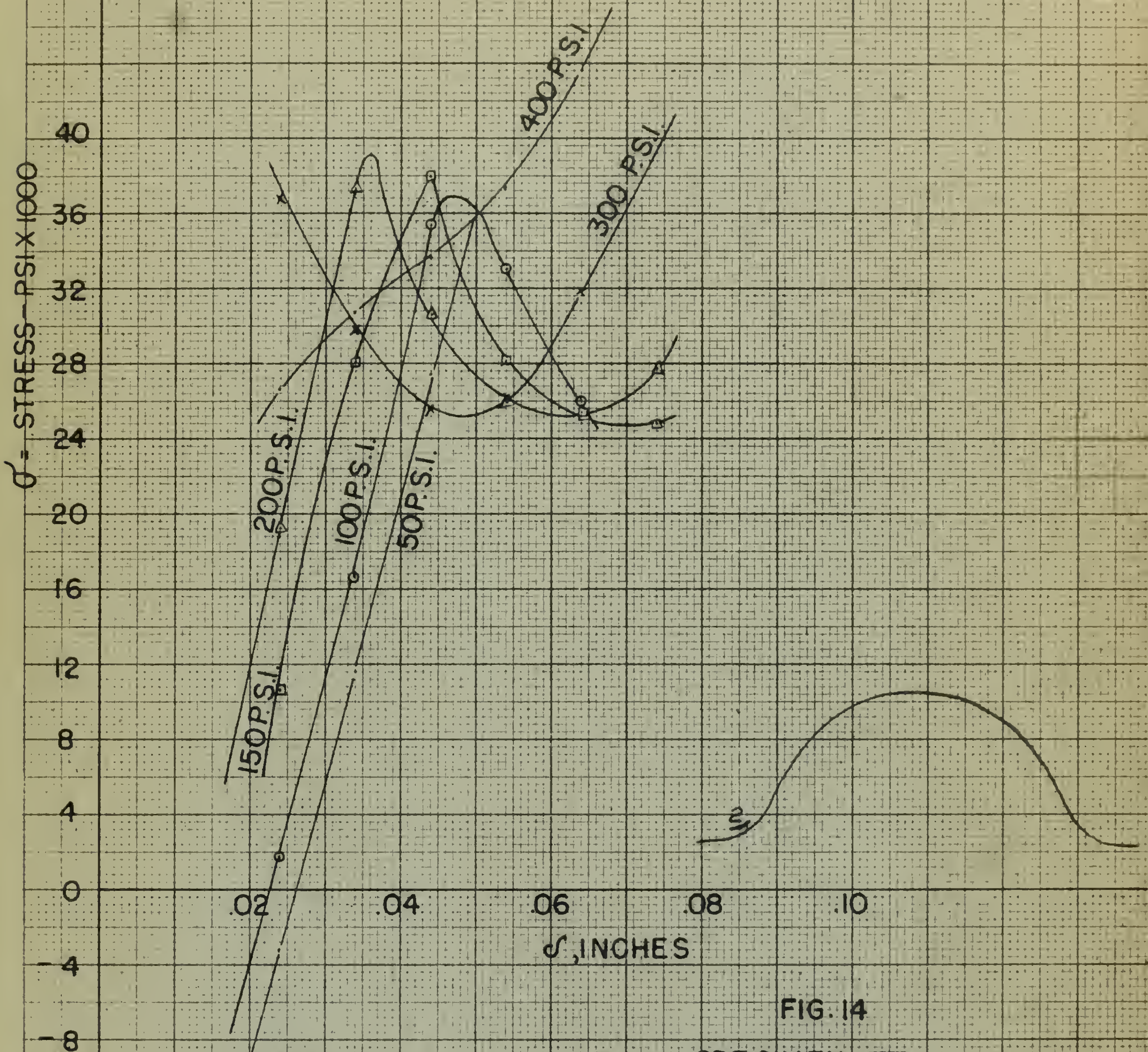



FIG. 14

SPECIMEN III

GAGE 2

δ VARIED AT CONST. INT. PRESS.

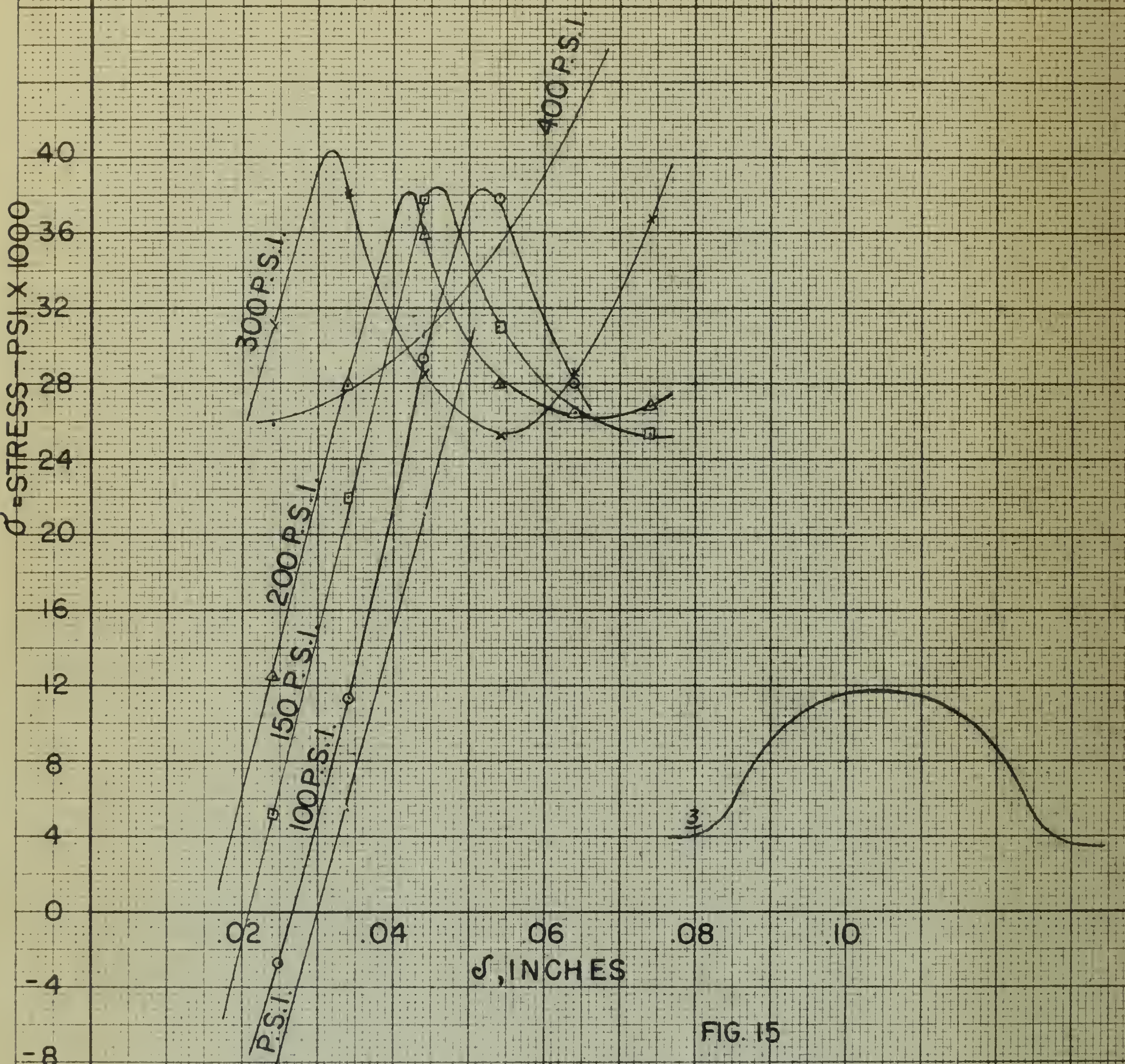



FIG. 15
SPECIMEN III
GAGE 3
δ VARIED AT CONST. INT. PRESS.

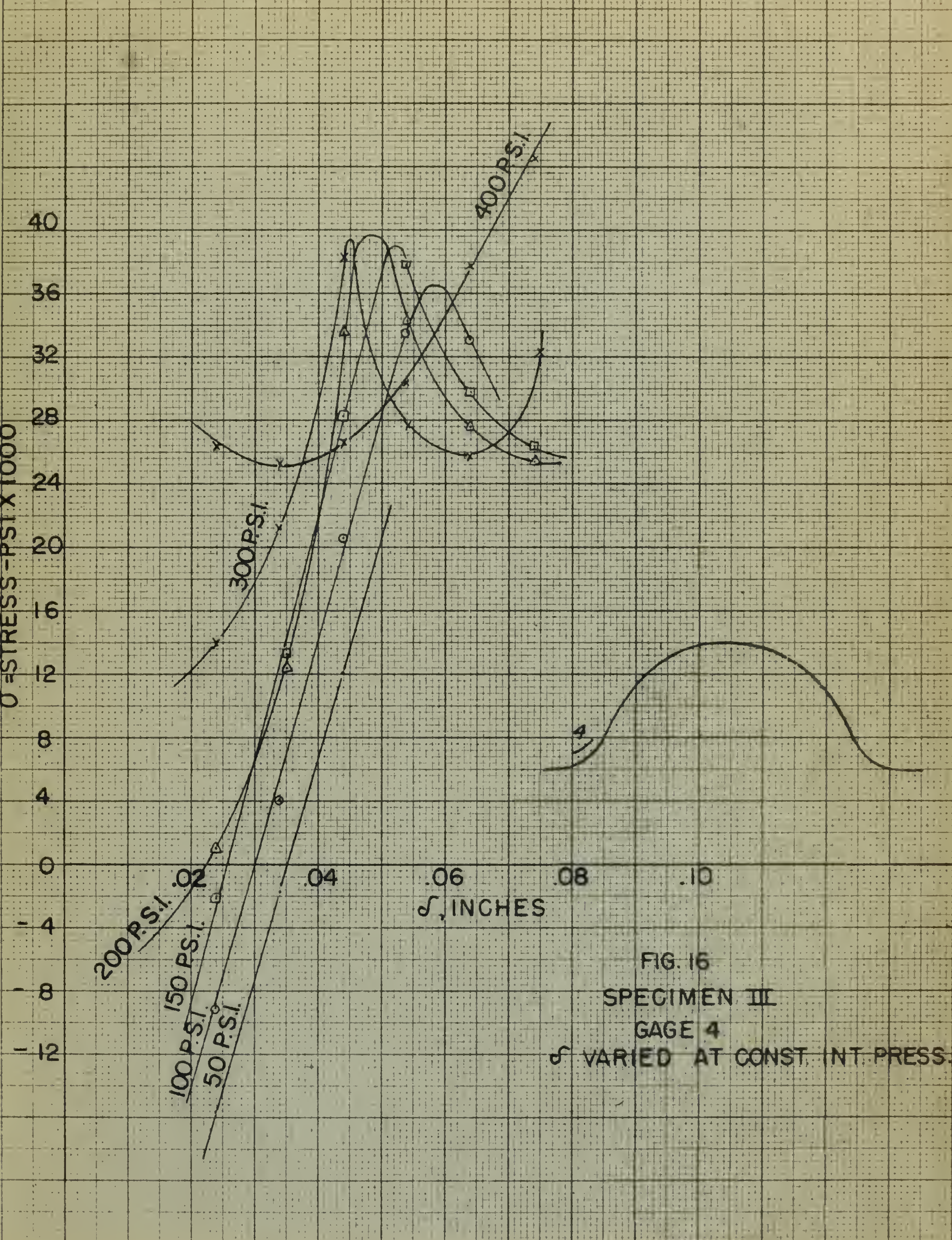



FIG. 16
SPECIMEN III
GAGE 4
δ VARIED AT CONST. INT. PRESS.

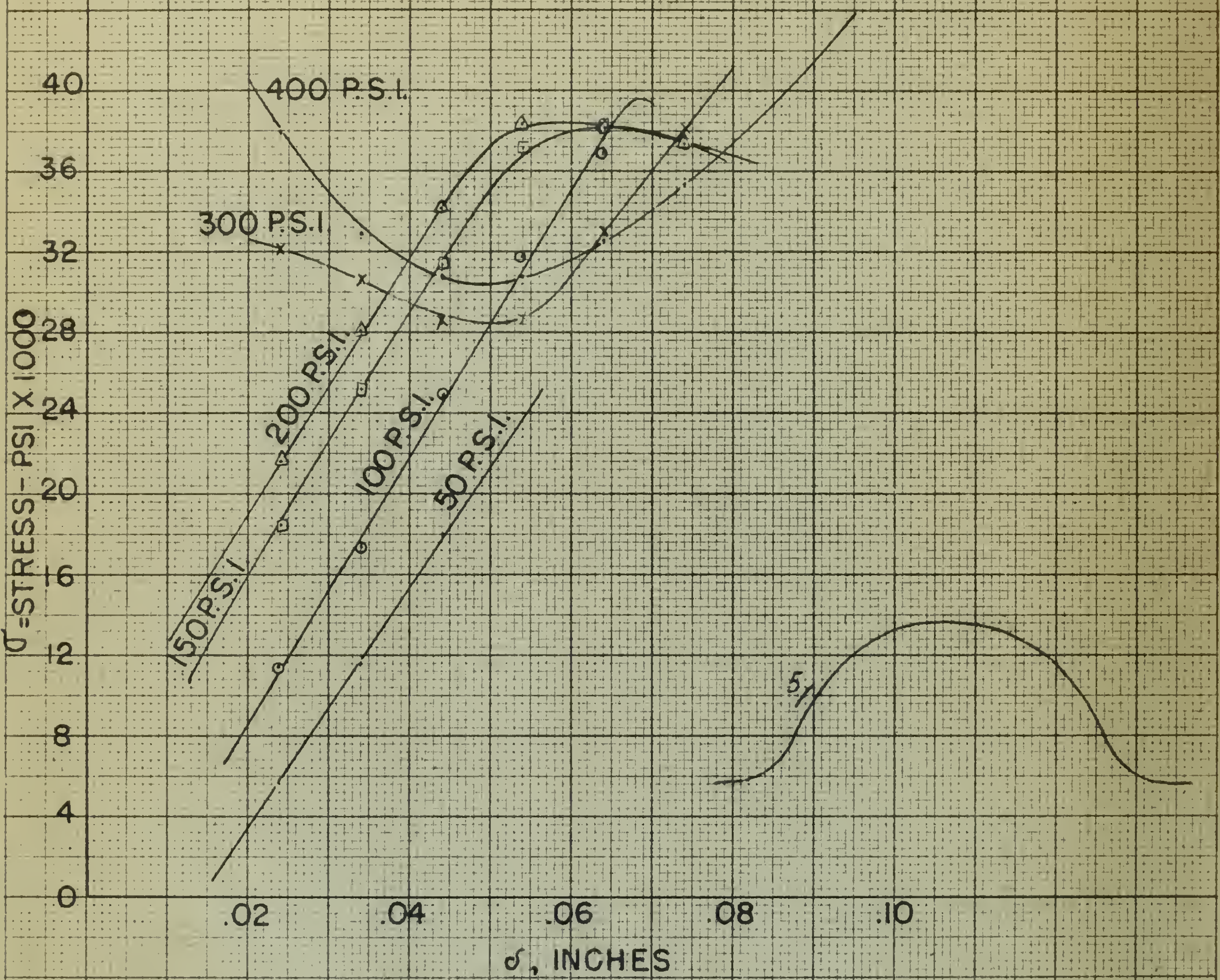



FIG. 17
SPECIMEN III
GAGE 5
δ VARIED AT CONST. INT. PRESS

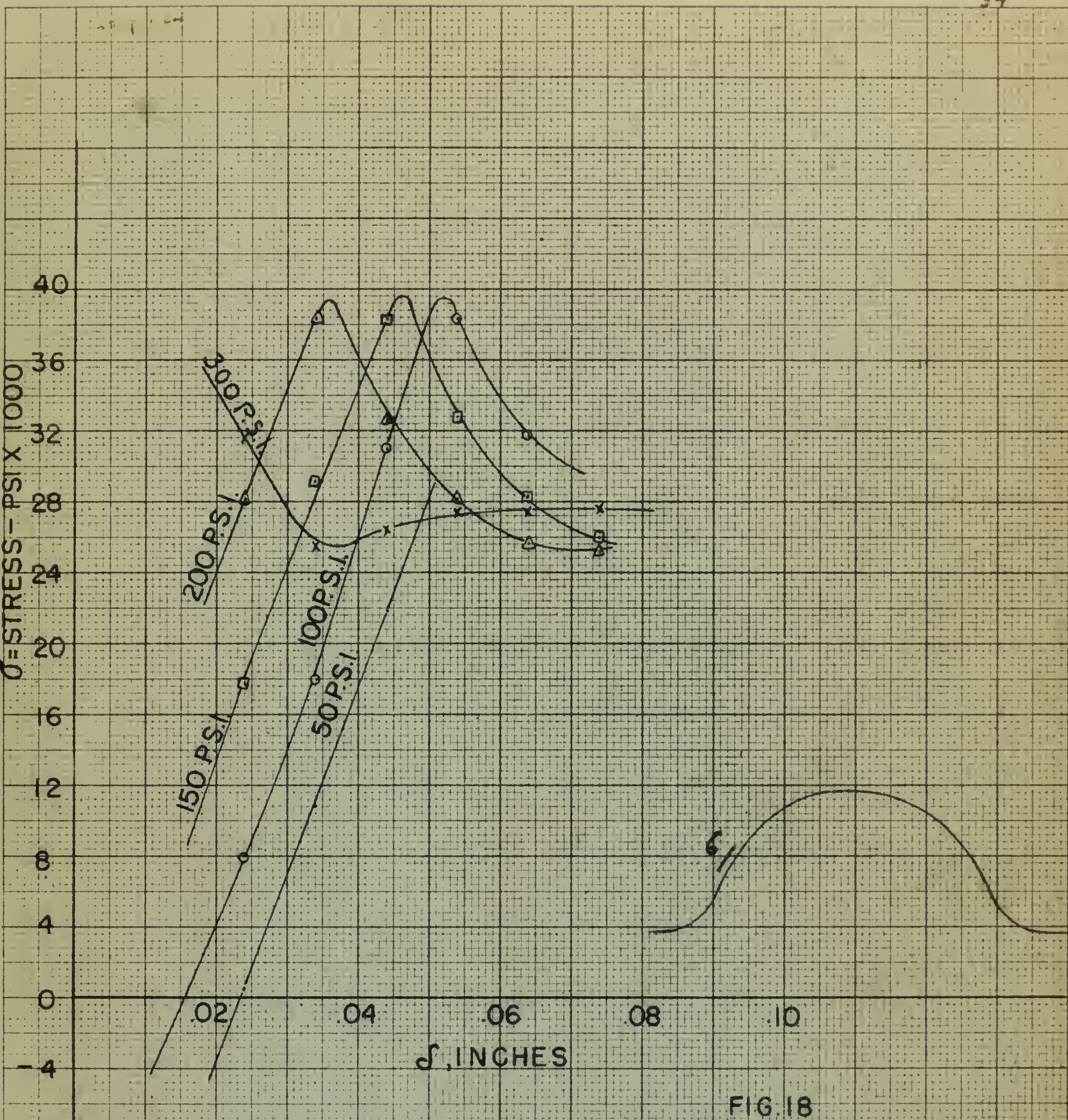



FIG. 18
SPECIMEN III
GAGE 6
δ VARIED AT CONST. INT. PRESS.

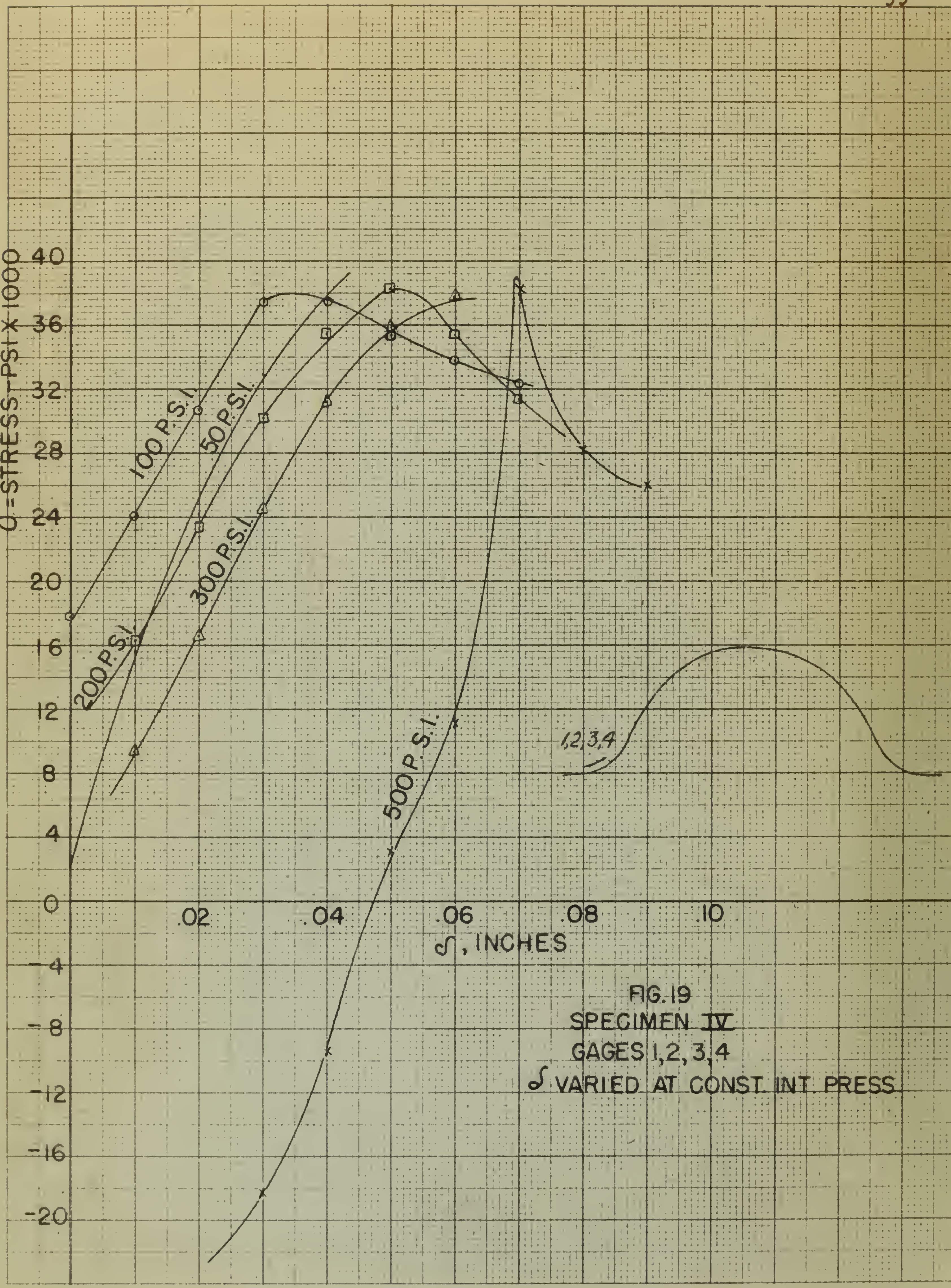



FIG. 19
SPECIMEN IV
GAGES 1, 2, 3, 4
δ VARIED AT CONST. INT. PRESS.

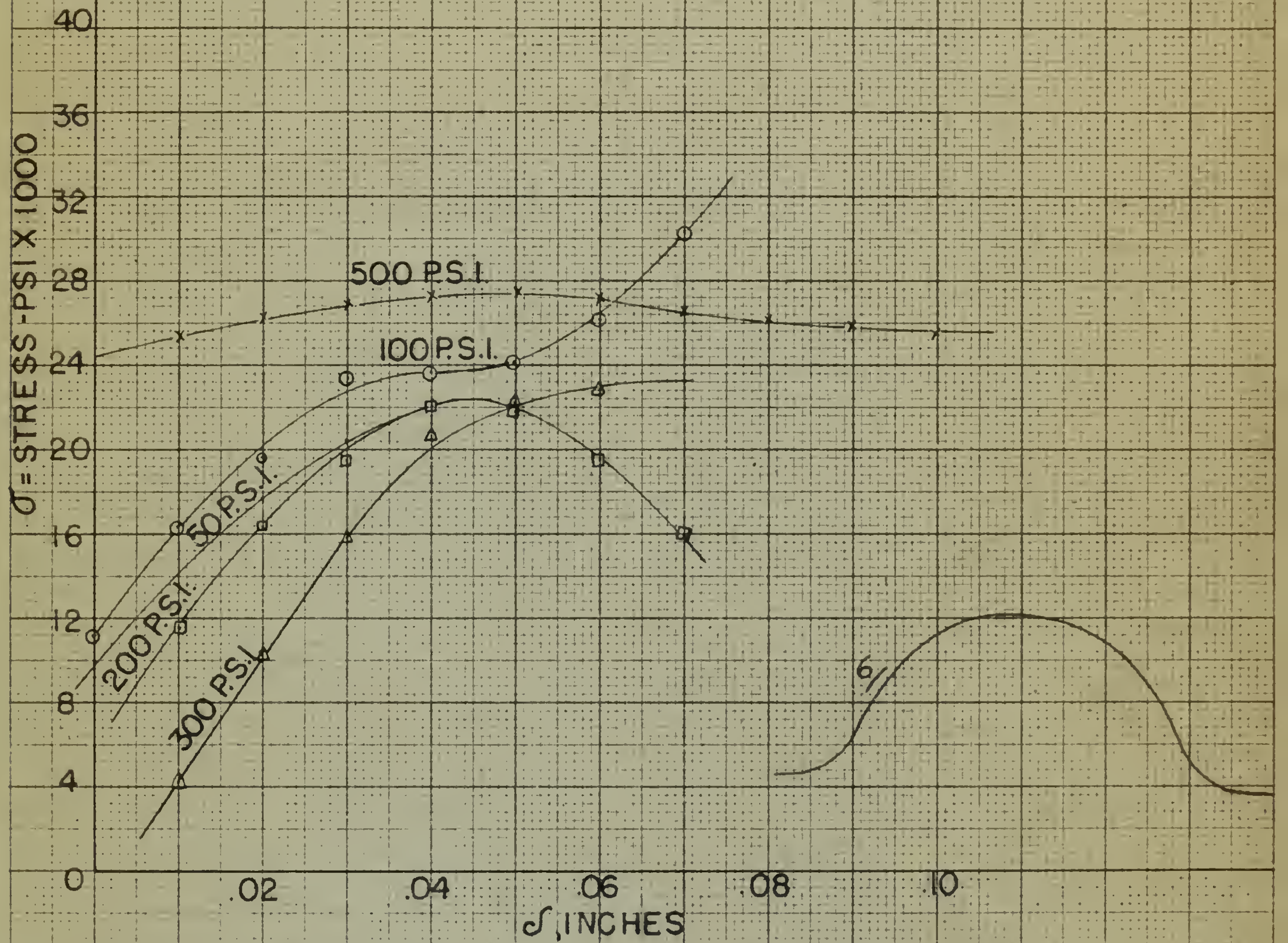
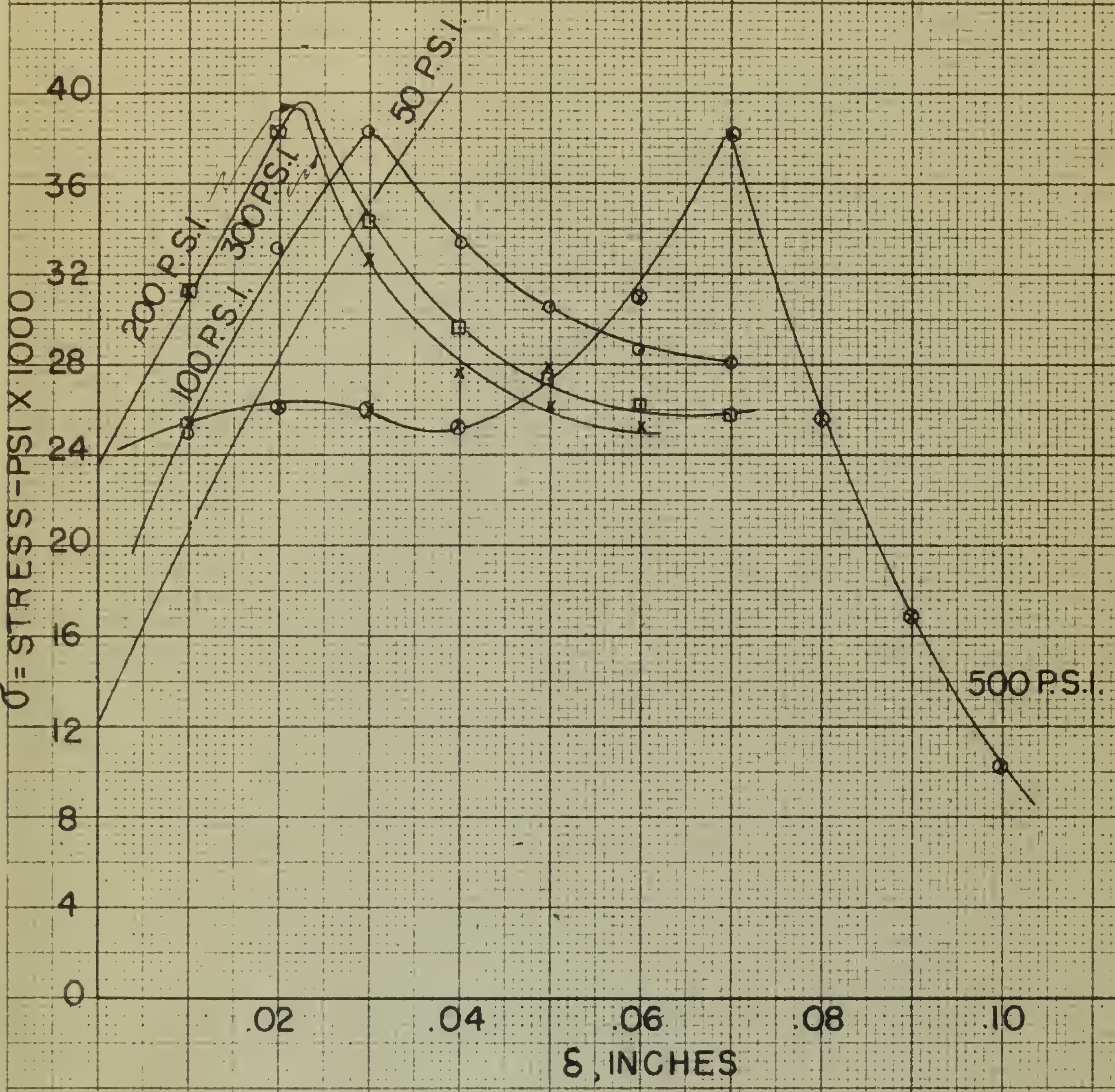



FIG. 20
SPECIMEN IV
GAGE 6
δ VARIED AT CONST. INT. PRESS.



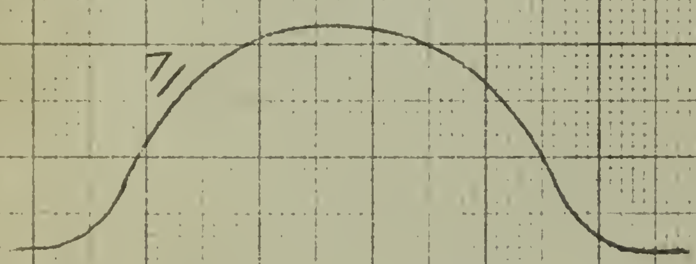




FIG. 21
SPECIMEN IV
GAGE 7
δ VARIED, INT. PRESS. HELD CONST.

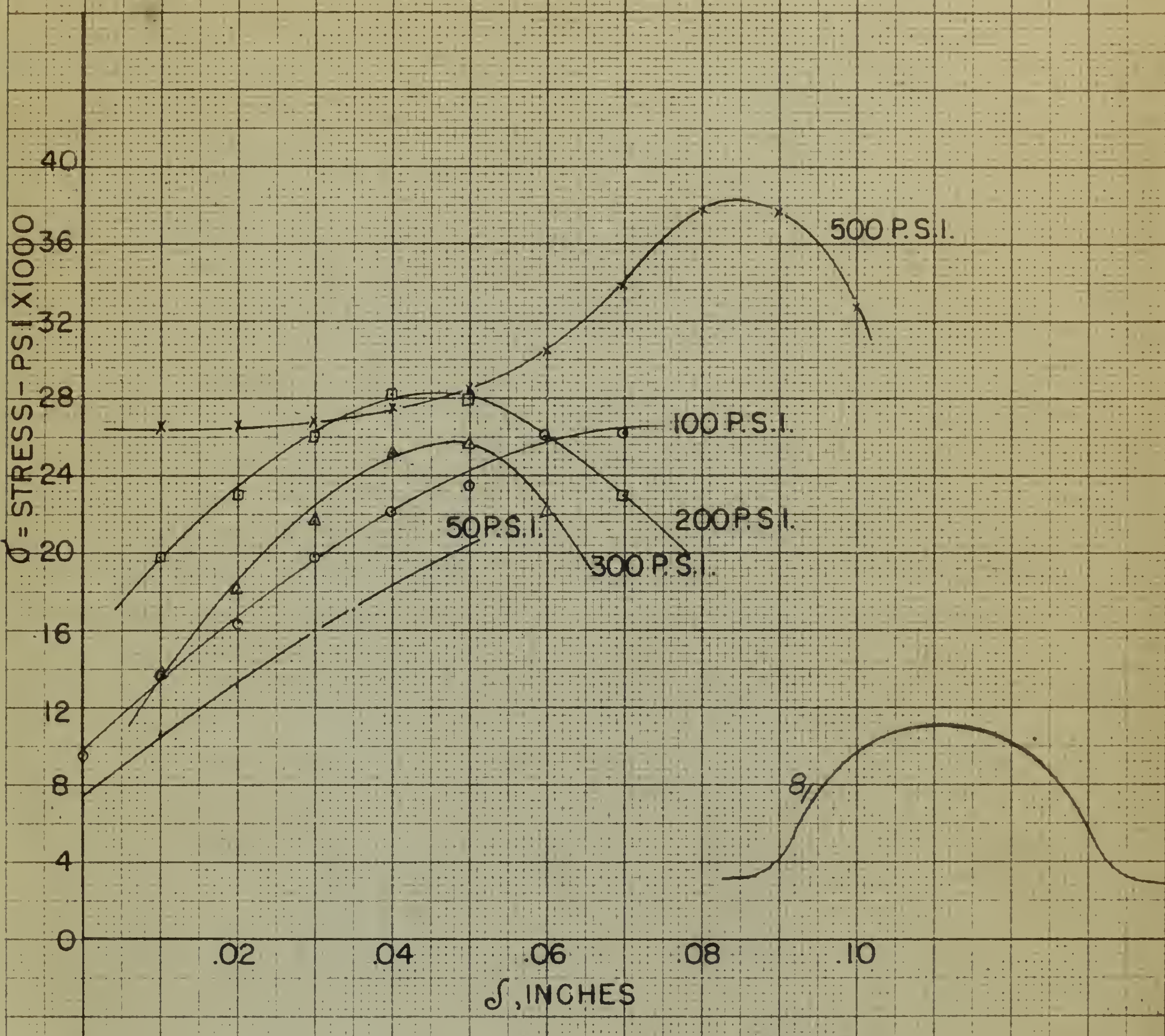



FIG. 22
SPECIMEN IV
GAGE 8
δ VARIED AS PRESS. HELD CONST.

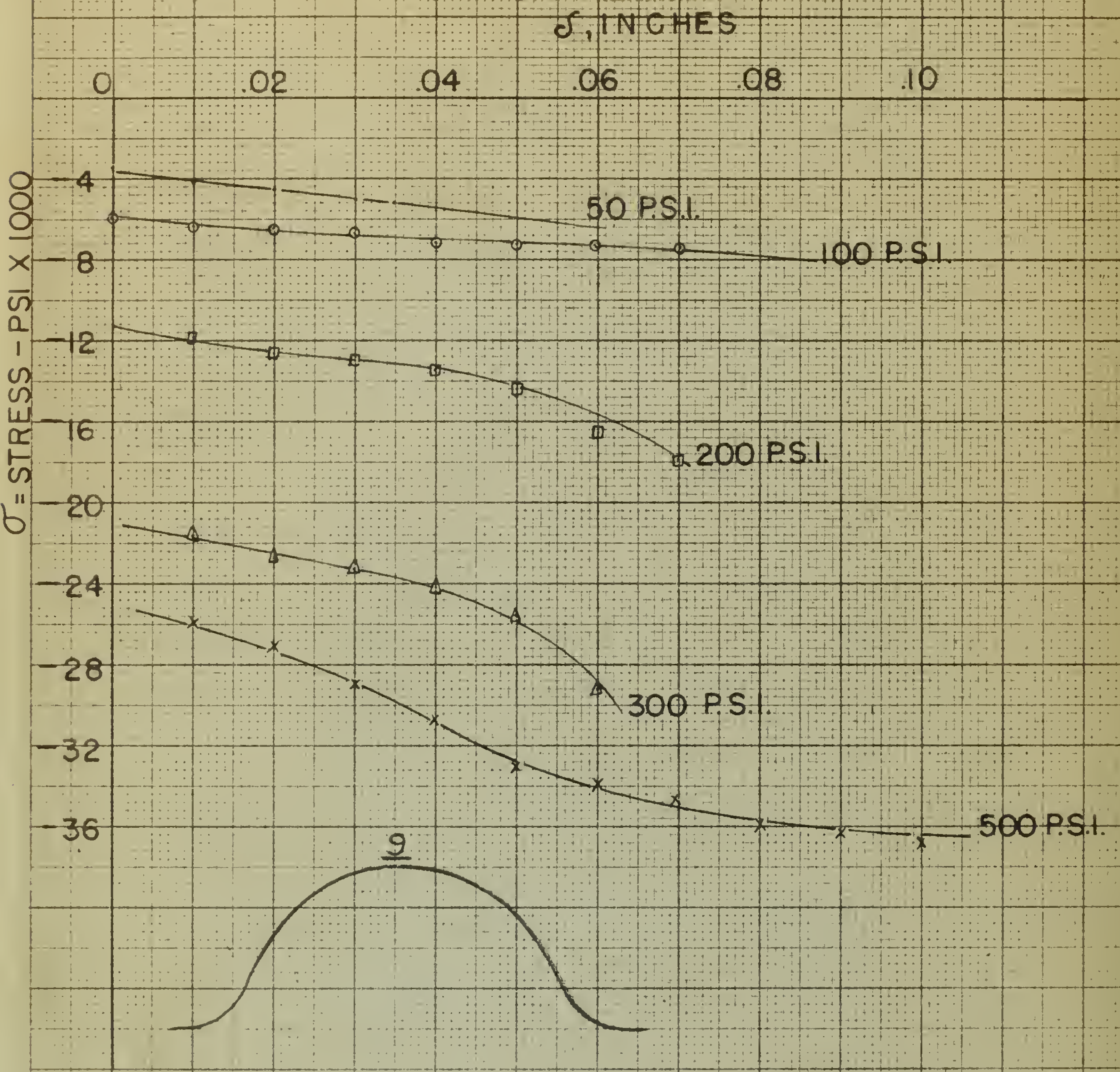



FIG. 23
SPECIMEN IV
GAGE 9
δ VARIED AS PRESS. HELD CONST.

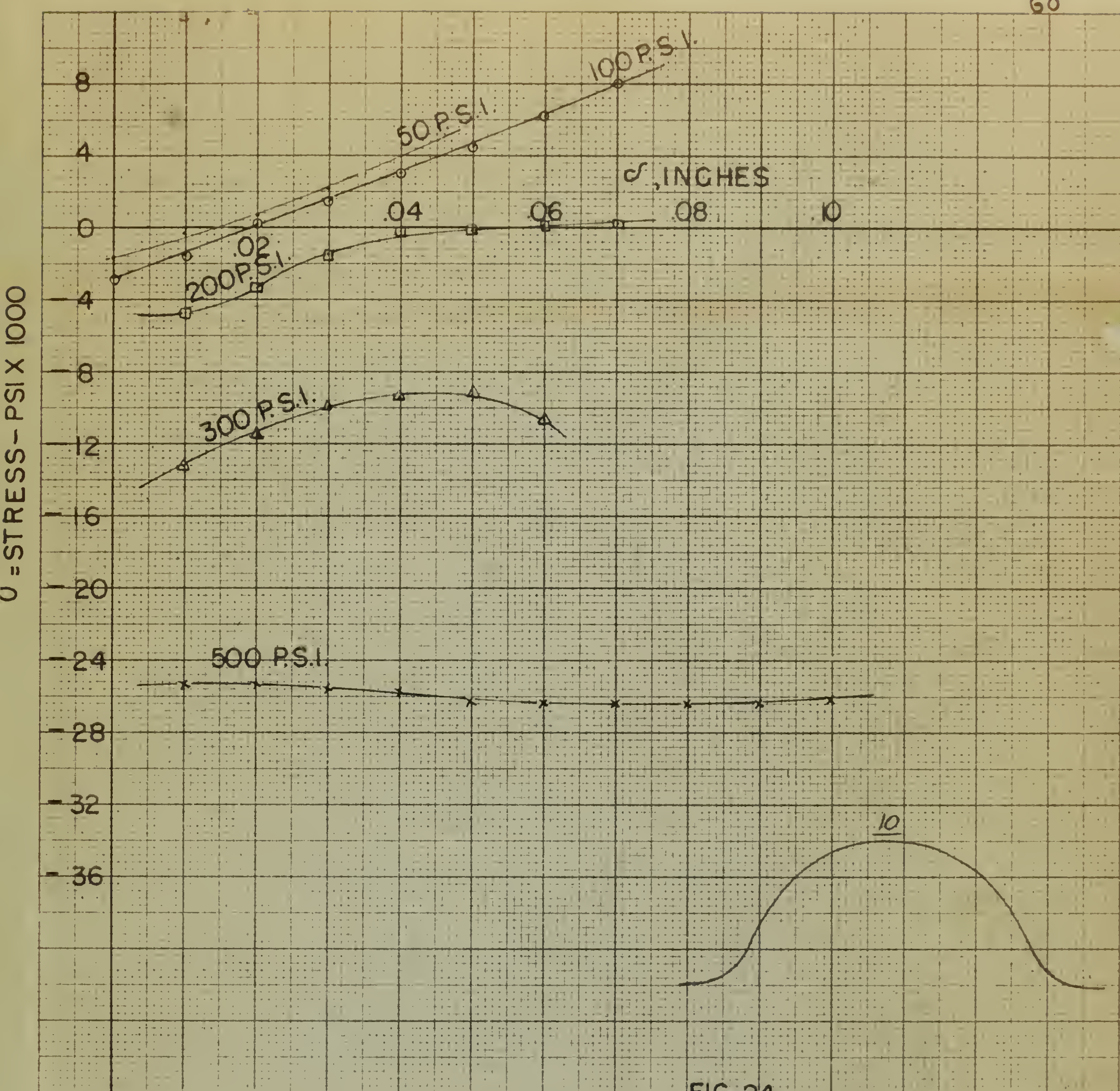
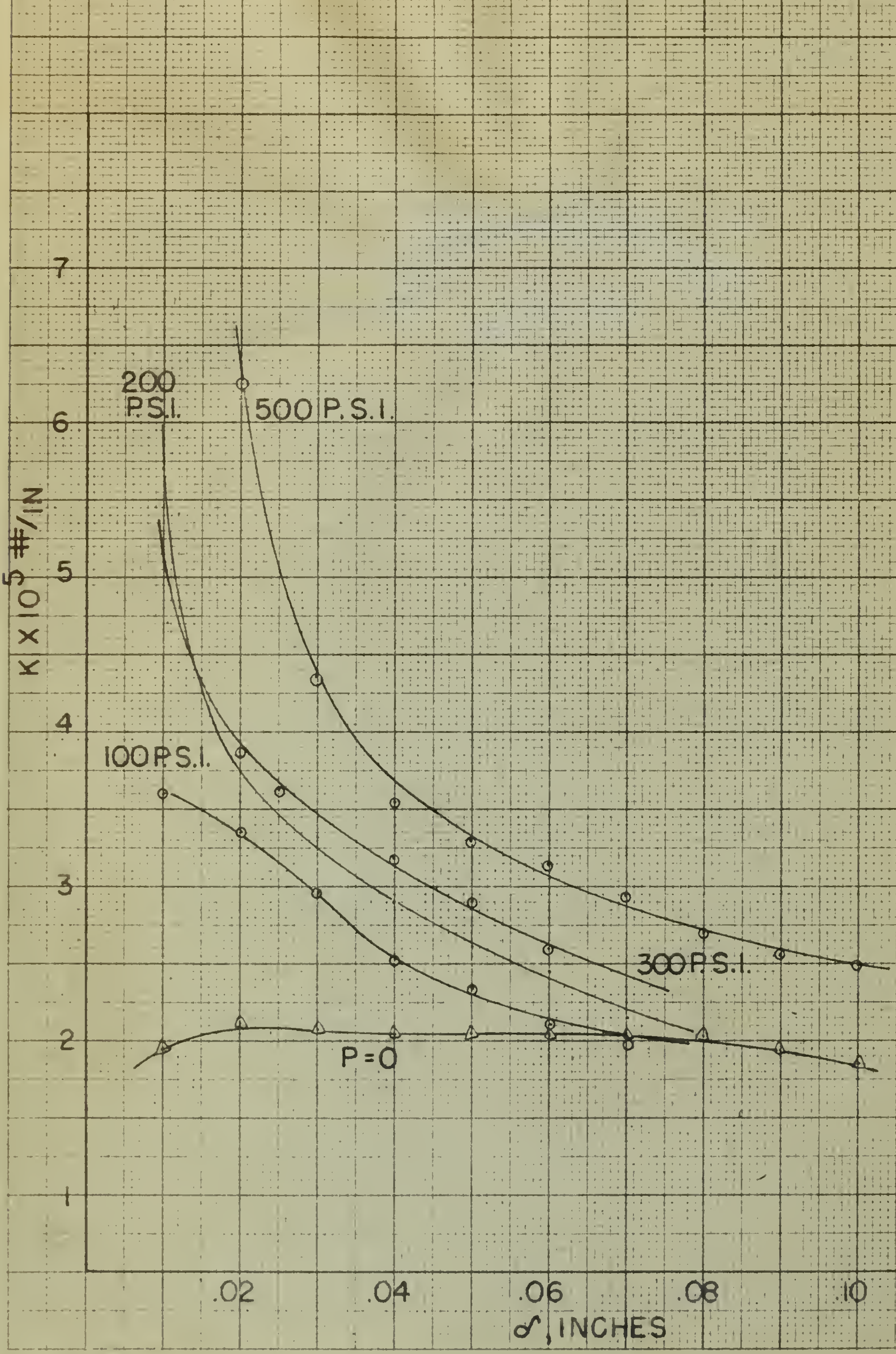



FIG. 24
SPECIMEN IV
GAGE 10
σ VARIED AS PRESS. HELD CONST.

FIG. 25



7
6
5
4
3
2
1
K X 10^5 #/IN
200 P.S.I.
500 P.S.I.
100 P.S.I.
300 P.S.I.
P = 0
.02
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.06
.08
.10
δ, INCHES

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